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CONSTRUCTION OF THE 18-METER WOODEN BLADE AT NIBE-MOELLE B.

Peter C. Nielsen

Translation of "Konstruktion af 18M Fritbaerende Traevinge til Moelle B i Nibe", Institutet for Bygningsteknik [Institute of Building Technology and Structural Engineering], Aalborg, Denmark, Report No. 8203, March 1982, 100 pp.

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16. Abstract

This report consists of one main report and 6 supplementary reports dealing with the construction of an 18 m cantilevered wooden rotor blade for wind turbine B at Nibe, Denmark. The main reports describes the project and draws the conclusions based on calculations and tests made on the rotor-blade. The supplementary reports furnishes details regarding choice of material, construction, attachments and fittings, stress problems and loads, etc.

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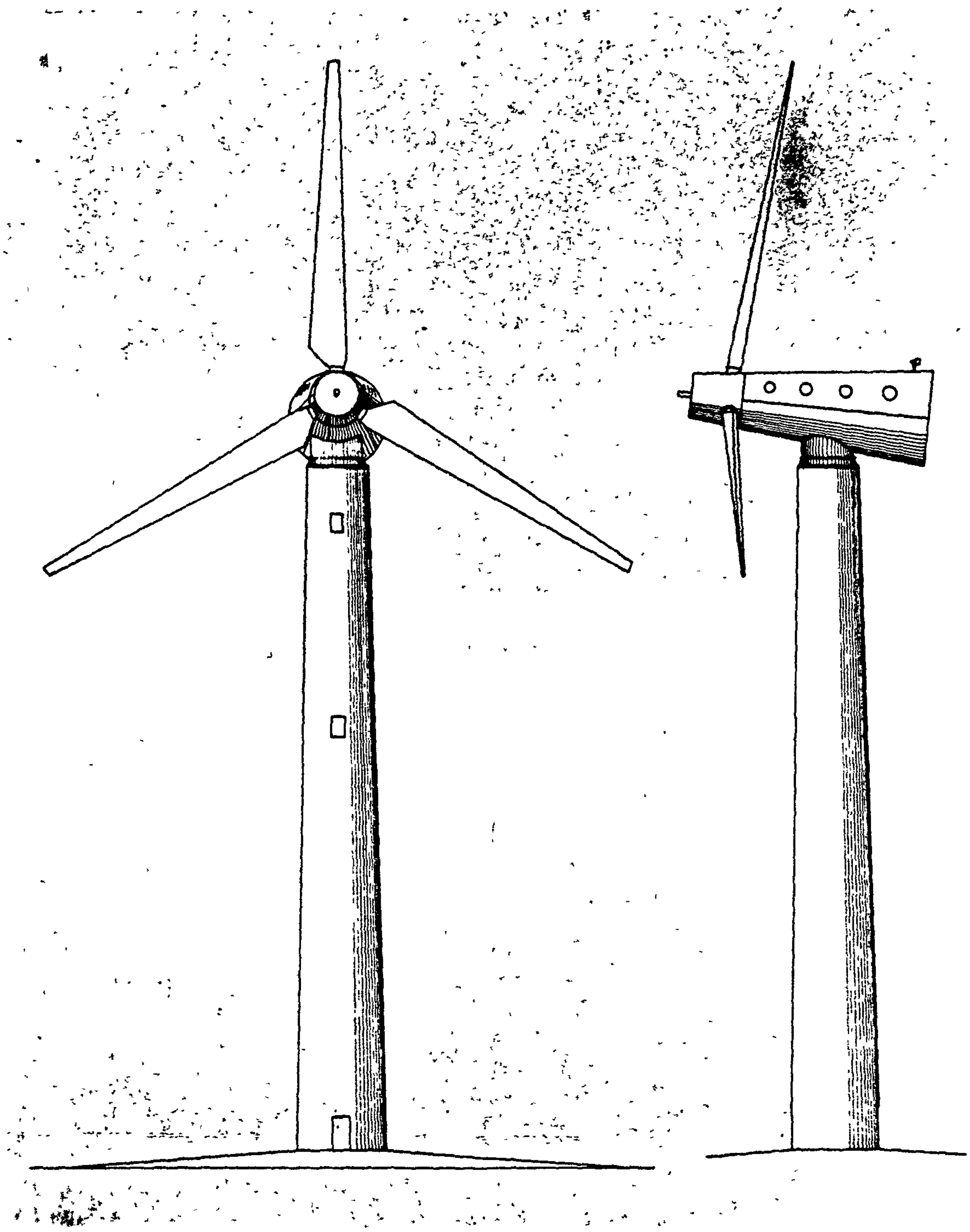

WOODEN ROTOR BLADES FOR WIND TURBINE B AT NIBE

* Numbers in the margin indicate pagination in the foreign text.

PREFACE

The Steering Committee of the Wind Power Program of the Department of Energy and of the Research Institute of the Danish Electric Power Companies selected in September of 1981 a team which was given the task of investigating the feasibility of using wooden rotor blades for large wind turbines with special attention to the experimental wind turbines at Nibe. /II

The Committee which has led the work on the present report was composed as follows:

- Henning Grastrup, ELSAM[1]
- Lars Pilegaard Hansen, Institute of Building Technology and Structural Engineering, Aalborg University Center (AUC)
- Preben Hoffmeyer, Department of Wood Technology, Technical Institute (TI)
- Ejgil Jensen, Research Institute of the Danish Electric Power Companies (DEFU)
- Dan Jepsen, LNJ Spaendtrae[1]
- Finn Johnsen, ELKRAFT[1] (Director)
- Hans Joergen Larsen
- Peter C. Nielsen, Institute of Building Technology and Structural Engineering, Aalborg University Center (AUC)
- Paul Nielsen, Research Institute of the Danish Electric Power Companies (DEFU)
- Hilmer Riberholt, Department of Load-bearing Constructions, the Danish Technical University (DtH)
- Stig Oeye, Department of Fluid Mechanics (AFM), the Danish Technical University.

Main Report:

Supplementary Reports:

[1] Translator's note: Name of Company.

REPORT I /cover p.

CONSTRUCTION OF AN 18-METER CANTILEVERED WOODEN ROTOR BLADE FOR WIND TURBINE B AT NIBE

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CONSTRUCTION OF AN 18-METER CANTILEVERED WOODEN ROTOR BLADE FOR WIND TURBINE B AT NIBE

by

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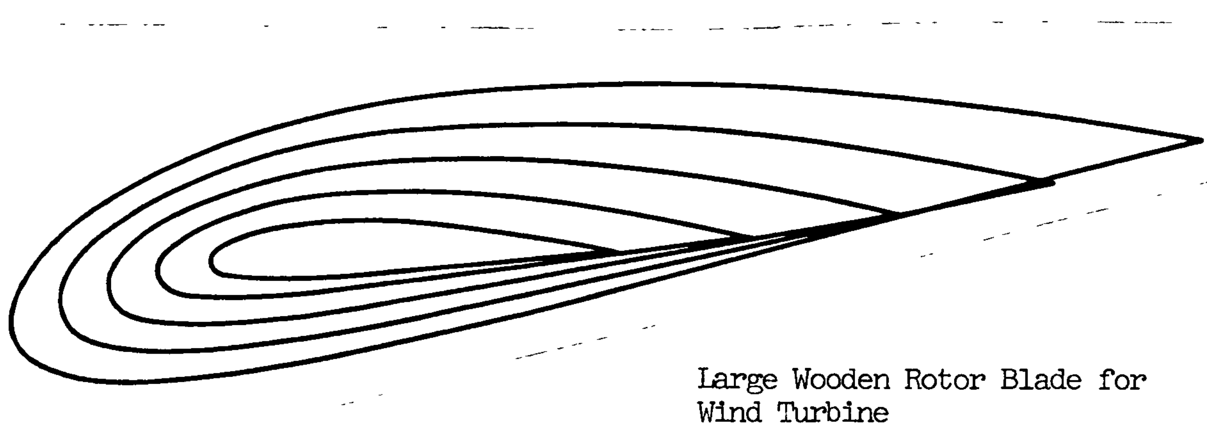

Large Wooden Rotor Blade for Wind Turbine

Translation of "Konstruktion of 18M Fritbaerende Traevinge til Moelle B i Nibe", Report no. 8203, ISSN 0105-7421, March 1982

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Summary /3

[In Danish = English summary]

Summary

This report describes an investigation into the possibilities of designing a wind turbine blade of laminated timber - glulam - and plywood. The aim has been to develop a rotor that can be fitted for one of the wind turbines built at Nibe by the Danish Ministry of Energy and the Association of Electric Utility Companies in Denmark.

A blade structure is suggested with the nose part made of solid glulam and tail panels made of plywood. These panels are carried on a system of plywood ribs.

The root of the rotor blade is fitted with a circle of tapered steel studs glued into the solid part of the blade to transfer loads to the rotor hub.

The calculations carried out show favorable static and dynamic behavior. A complete wooden blade is lighter than the existing rotor blade made of steel and glassfiber reinforced polyester.

1. INTRODUCTION /4

This report concerns an investigation into the feasibility of constructing a wooden rotor blade for use with the large-size wind turbines having an horizontal axle. The investigation was conducted for and financed by the Department of Energy and the Wind Power Program of the Danish Electric Power Companies (DEFU).

2. CONCLUSIONS

The investigations made revealed that:

- it is feasible to construct a wooden rotor blade of laminated wood and plywood which will be capable of satisfying the requirements regarding weight, strength and inherent vibration conditions;
- the matter of attachment, i.e. the fitting of the blade to the rotor hub, is most efficiently solved by means of a circle of glued-in stud bolts;
- the production of the blades puts stringent demands on the choice of materials and the laminating operations;
- experimental research should be conducted before the final construction of the rotor blade is decided upon.

3. PREREQUISITES

The starting point for the work on the development of wooden rotor blades for large-size wind turbines is the construction of a set of experimental blades for the wind turbine B at Nibe under the auspices of the Department of Energy and of the Danish Electric Power Companies. These blades must have the same dimensions as the existing glassfiber/steel blades. However the distribution of bulk at the root end may be varied. The intention is to construct an 18 meter long blade with an NACA 44 xx profile and 11° torsion of the wooden chord from tip to root; cf. Figure 1.

The wooden blade including its fittings must not weigh more than 3000 kilograms.

The placement of the point of gravity at the leading edge of the profile necessitates that the original idea of a solid wooden blade (cf.[10]) must be abandoned in favor of a hollow construction made of laminated wood and plywood.

4. REVIEW OF THE LITERATURE /6

There exists some literature on the use of wood for small wind turbines with horizontal as well as with vertical axles; see for instance [4], [8] and [12].

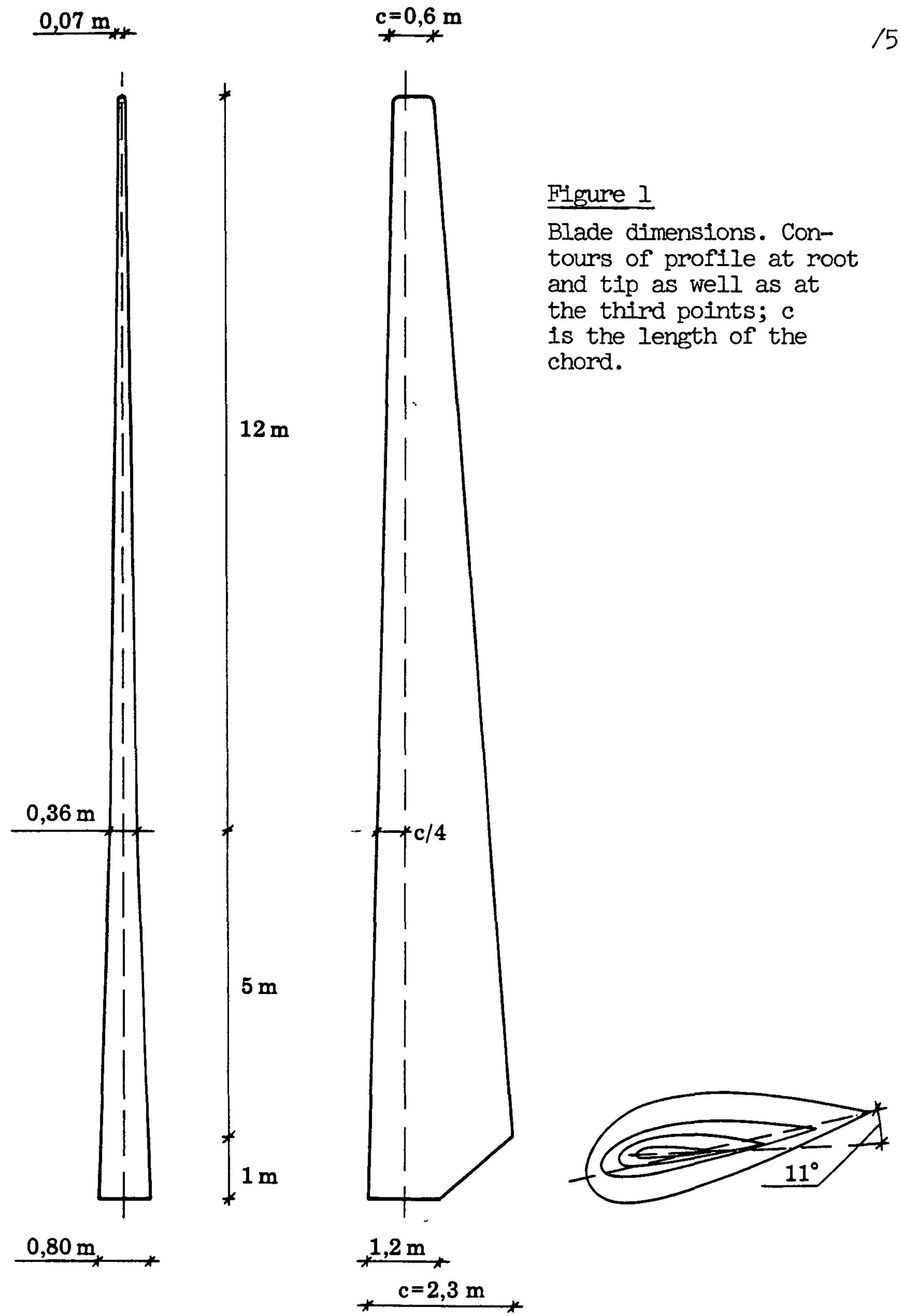


Figure 1

Blade dimensions. Contours of profile at root and tip as well as at the third points; c is the length of the chord.

The only reference we have been able to find regarding large wooden rotor blades concerns a set of blades for wind turbine model MOD-0 A developed for the wind power program in the United States of America. These blades, 18 meters in length, are constructed of plywood glued together with epoxy resins.

There is in addition a paper about the construction of wooden turbine blades for a wind tunnel.

The individual references are briefly reviewed below:

Gougeon and Zutech [7] described the development of the 18 m long MOD-0 A wooden blade, the basic idea of which is a profile constructed of thin layers of plywood glued together with epoxy resins under a very moderate pressure. The advantage of this method seems to be that great resistance to humidity is achieved since the entire profile is impregnated with epoxy resins which prevent the penetration of moisture and of vapors. The attachment is accomplished by means of a set of glued-in studs.

Gougeon and Thomes [6] furnished in a preliminary report on the MOD-0 A blades a detailed description of the manufacturing procedures developed. Briefly this report stated that it aimed at constructing two sides of the blade, an upper and a lower half, each in its own matrix. The pressure was applied by means of a vacuum jacket. When the two halves had been trimmed, they were glued together while still sitting each in its own matrix. This report describes also the basis for the calculations used for the projections.

Faddoul [5] has written a report containing a review of the static and dynamic tests made on the MOD-0 A wooden blade following its construction. The test series with glued-in studs leading to the final construction of the attachment are described. He also reported on the full scale tests with a 6 meter root end constructed in order to test the solution of the attachment.

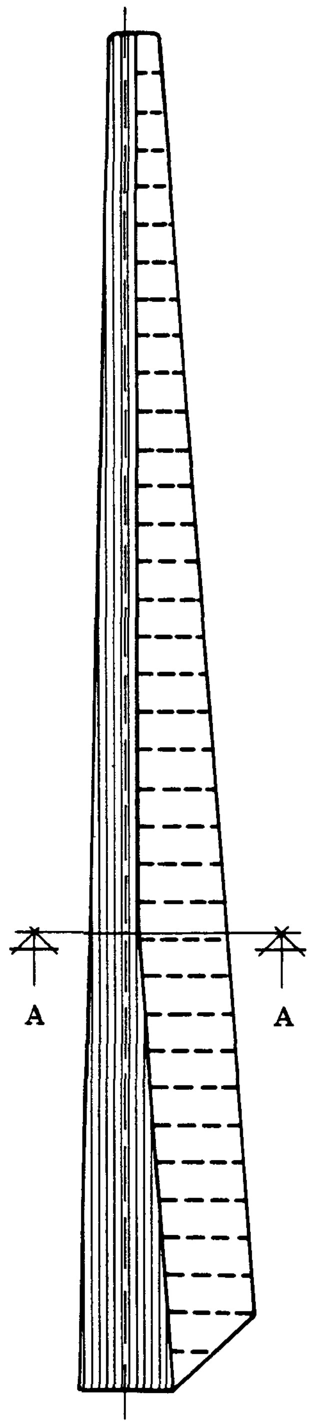
Finally a paper [11] shall be mentioned which describes the construction of a wind tunnel rotor with 6 blades made of Canadian spruce wood glued together with resorcinol glue. The rotor blade is 3.8 m long and has chords varying from 1 to 1.22 m. The attachment is made by means of studs between two steel cover plates. The rotor blade which is solid was glued in two steps. First the smaller blocks consisting of 11 laminating boards, 19 mm wide, were built. The direction of the fibers of the individual boards formed an 11° angle toward the center line, alternating from side to side. The individual boards where weighted so that all the rotor blades had identical distribution of mass. Only wood free of knots or with very small knots was used.

Four laminated blocks were required for each rotor blade. These were glued together whereafter final trimming took place.

5. CONSTRUCTION OF THE ROTOR BLADE /7

From the beginning of the construction phase it was obvious that the wooden blade must be built as a hollow construction with a laminated wood beam at the leading edge of the blade profile. This beam should, thus, have an approximately D-shaped cross section. The desired dimensions are accomplished by gluing plywood panels to the trailing edge of the laminated beam.

These panels must be supported by a row of ribs forming the rear part of the blade profile. In respect to the torsion rigidity and the dominating dynamic stress on the construction to which the blad is subjected it must be built so that the laminated wood and the plywood function in unison. This assures that the blade will act as a monolithic piece of construction. The segmental construction is illustrated in Figure 2.



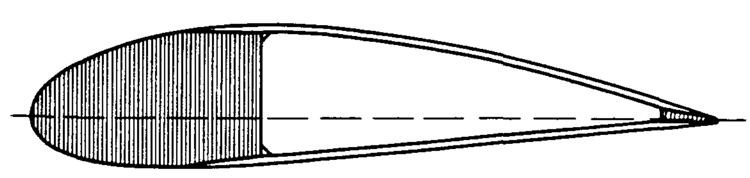
5.1 Concerning Materials

It is planned to make the laminated wood part of thin boards of high quality soft wood, for instance Norway spruce grown in Jutland [13]. Because of the distribution of mass and the rigidity of the blade it is necessary to determine the weight and the rigidity of the individual boards in advance and to sort them accordingly into different categories. Determination of rigidity can possibly be done at an automatic grading plant. The material for the individual blades is thereafter selected from the different categories of boards by means of templates.

The plywood to be used can consist of ordinary construction plywood but it is advantageous to use special plywood, for instance such made of birch. The special plywood is in addition often available in longer panels so that the number of splices between the plywood panels in the finished upper and lower walls can be reduced.

Water and boil-proof resorcinol glue should be used for gluing together the solid supporting beam. All other assembling of the blade can be done with epoxy resins and fillers.

The surface treatment of the finished blade can be done with pigmented acrylic-enhanced



Section A - A

Figure 2. Rotor Blade constructed of laminated wood and plywood. /8

polyurethane. As described in [6] the surface can be treated with an epoxy material as well. The surface treatment must be espcially carefully done where end wood is exposed.

5.2 Concerning Construction

The supporting beam made of laminated wood should be constructed of edge-glued wedge-shaped laminating boards, all with parallel fibers. The wedge shape can also be achieved by cross- cutting and cross-layering segments of a constant width. The fibers of one board will then form an angle toward the fibers of the neigboring board. The decision regarding the possible use of cross-cutting must await must await the results of experimental research; cf. Appendix D.

The tapering of the beam at right angle to the chord of the blade profile can be achieved by means of wedge-shaping the boards as mentioned above. The tapering in parallel with the chord is achieved by gradually stepping of the beam which means that the length of the boards will vary. During the gluing of the beam it must be twisted around its longitudinal axis in relation to the angle between the chords at the tip and at the root end of the finished blade. The individual boards should preferably be joined by means of diagonal splices which have better fatigue strength than fingering splices.

The manufacturing of the solid part made of laminated wood can also be done according to the principles developed by LNJ Spaendtrae. The D-beam is glued together of two halves along a surface parallel to the chords of the blade profile and the longitudinal axis of the beam. Each half-beam is glued together of separate boards in a mould where pressure can be applied. This mould is shaped so that the least possible spill of wood occurs during the final trimming of the D-beam. The method is described by Jepsen in [15].

When constructing the first set of blades the shaping of the laminated /9
wood into the D-shape desired must be done by hand using templates cut according to computer designed profile contours. Later an arrangement should be developed so that the manufacturing can be done automatically.

When the beam of laminated wood has obtained its final shape the precut plywood ribs intended for support of the plywood panels shall be glued on. This work moment and the following application of the plywood panels to the upper and lower sides of the blade can most likely be performed in a fixed working arrangement.

Both the plywood panels, together forming the trailing part of the blade profile, must be shaped before being attached to the part of laminated wood. The individual portions must be joined by diagonal splices, whereafter the panels can be cut to final shape. After hardening and the installation of wooden splints along the rear edge, the upper panel is glued on. A special clamping device for gluing on the two panels is being developed.

After mounting the attachment fittings and the end pieces which close the blade cavity, trimming and surface treatment shall follow.

Final decisions regarding the manufacturing procedures must be made in consultation with the technicians at the laminated wood factories concerned.

5.3 Details Regarding the Mounting

A report will be given regarding the planned development of fittings for mounting the blade under discussion.

The splicing of the laminated wood and the plywood must be done in the form of diagonal splices where the diagonal cut in the plywood shall face the laminated wood. The two plywood panels shall be joined along the rear edge of the blade by being glued to a wooden strip.

The blade cavity shall be closed at the root end and at the tip by means of grooved veneer panels. At the tipe of the laminated wood part holes may be taken out where balancing weights can be attached before the end plate is adapted.

The ribs over which the plywood panels are stretched shall be attached to the laminated part of the wood by being glued in between triangular wooden strips.

Sketches of the individual solutions of these details are illustrated in Appendix A.

6. CALCULATIONS CONCERNING THE ROTOR BLADE /10

In connection with the decisions regarding the construction, a preliminary stress analysis was made. It is reproduced in section 6.2 A later analysis was based on dynamic calculations made on the forced response of the blade. This is found in section 6.3. The vibration analysis is reported in section 6.4.

6.1 Basic Conditions for the Calculations

It is assumed that the blade shall be considered as a rigidly attached cantilevered beam.

The cross section constants vary from section to section. In order to simplify the calculations the blade was divided into 1 m segments with a constant diameter. The load on the blade is applied in the form of single stress at the nodes between the individual blade segments. This method has previously been used by Petersen [11].

All calculations shall be made while using a coordinate system with the x-axis in parallel with the chord of the blade profile, the y-axis at right angle to the chord and the z-axis along the systemic line of the blade; cf. Figure 6.

A special computer program was developed for calculating the cross section constants. This enables us to make a numerical determination of these at any optional cross section throughout the blade.

The routine calculations of the coordinates of the profile contours is borrowed from the Profile Plotting Computer Program PROP [3].

This program divides the cross section into small rectangular elemental areas. The areas, the static moments and the inertial moments are obtained by summation of the contributions of the individual elements. The difference in elasticity module (E-module) between wood and plywood is included by calculating with transformed cross sections. The program is illustrated in Appendix B.

An idea of the size of the torsion inertia moment which is considered to be of secondary importance is obtained by calculating this moment for both a solid and a hollow thin-shelled cross section shaped like the blade profile. The mean of both these data is used.

The following parameters of the materials of laminated wood are used for the calculations:

Density	$p = 450$ kg/m^3
Elasticity module	E = 13.5 GPa
Dislocation module	G = 1.0 GPa
Typical bending strength	$f_k = 50$ MPa

These data are determined in relation to carefully selected material.

The relationship between the weight of plywood and that of laminated wood is set at 1.2 while the corresponding relation between the E-modules has been set at 0.8. It is furthermore assumed that there shall be no essential differences in strength between the plywood and the laminated wood.

The theory concerning homogenous isotropes and linearly elastic materials constitutes the basis for the calculations.

The following calculations have been made on a blade profile the leading third of which consists of solid laminated wood and the trailing part of which is formed of two 12 mm thick plywood panels. Cross section data on this blade have been calculated by means of the computer program described and included in Appendix C.

6.2 Stress Analysis: Preliminary Calculations /11

A preliminary calculation for determining the principal shape of the blade was made on the basis of the maximally permissible aerodynamic loads on the blade.

Extreme stress during a number of load moments was made the basis for the stress analysis of the blade construction. The load moments investigated were determined in consultation with the Department of Fluid Mechanics (AFM) at the Danish Technical University (DtH).

The blades are affected by aerodynamic loads, their own weight and centrifugal forces. The latter have a component, the conic force, at right angle to

to the axle of the blade since this forms a 6^{o} angle with the rotor plane.

The Department of Fluid Mechanics has calculated the aerodynamic load on the blade under four conditions forming the basis for the load moments to be investigated. These loads are illustrated in Figure 3.

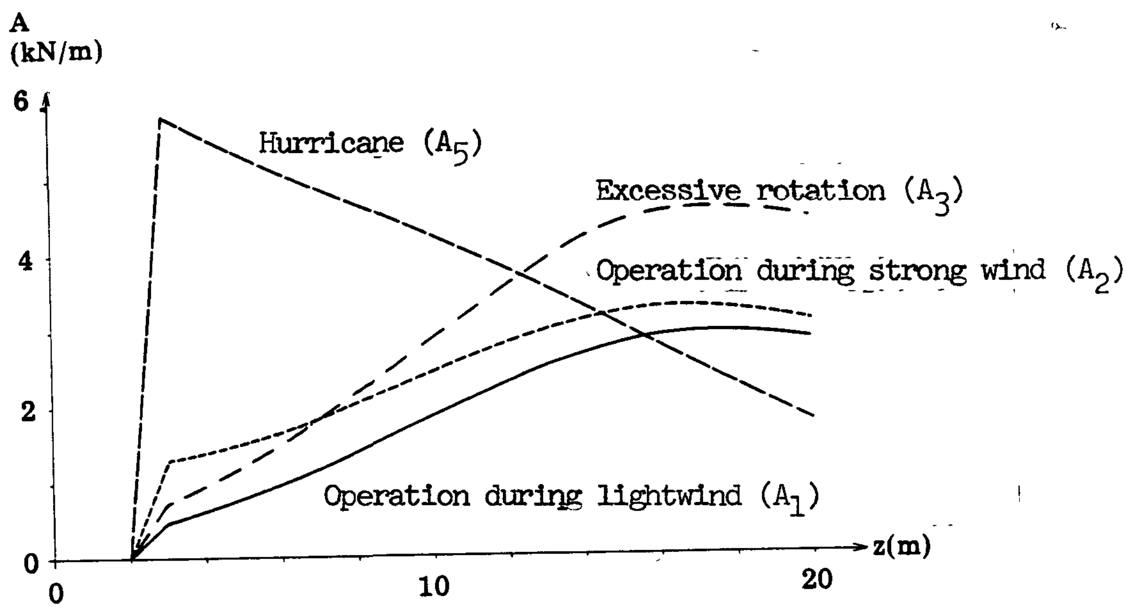


Figure 3. Linear load caused by wind at right angle to the blade axle.

The conic force and the normal thrust have been calculated to 3.5 rad/sec for the shape selected of the blade. The inherent weight and the conic force are illustrated in Figure 4. The variation in normal thrust is illustrated in Figure 5. /12

The load moments investigated are:

1. Operation during light wind
2. Operation during strong wind
3. 25% excessive rotation
4. Stopping following excessive rotation
5. During hurricane when not operating

A review of the contributions of the individual loads and their angular impact is furnished in Table 1 and Figure 6.

The dislocation forces and moments during the 5 load moments were calculated for the cross sections with a mutual distance of 1 m along the entire span of the blade. Those moments around the weak axis of the blade cross section which are of dominating importance for the magnitude of the stress appearing are illustrated in Figure 7. The sectional forces at the root end of the blade are reproduced in Appendix C.

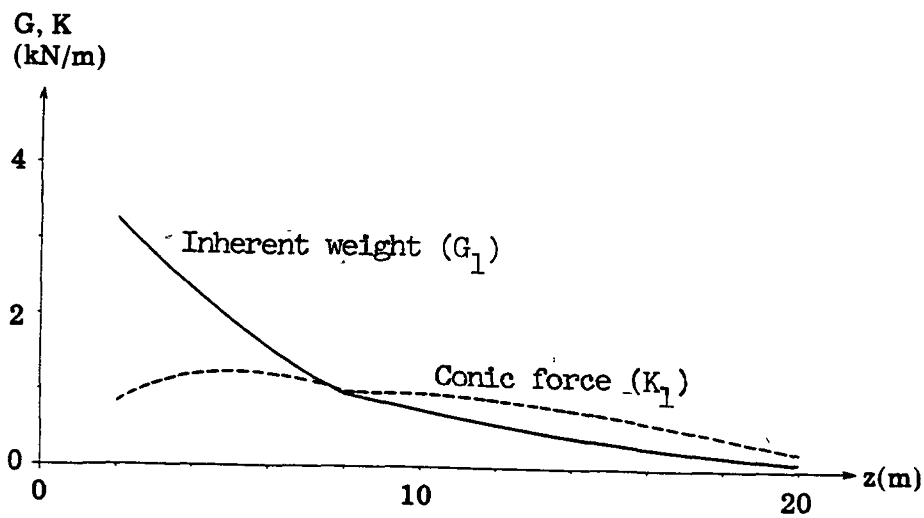


Figure 4. Linear load caused by inherent weight and conic force at right angle to the blade. /11

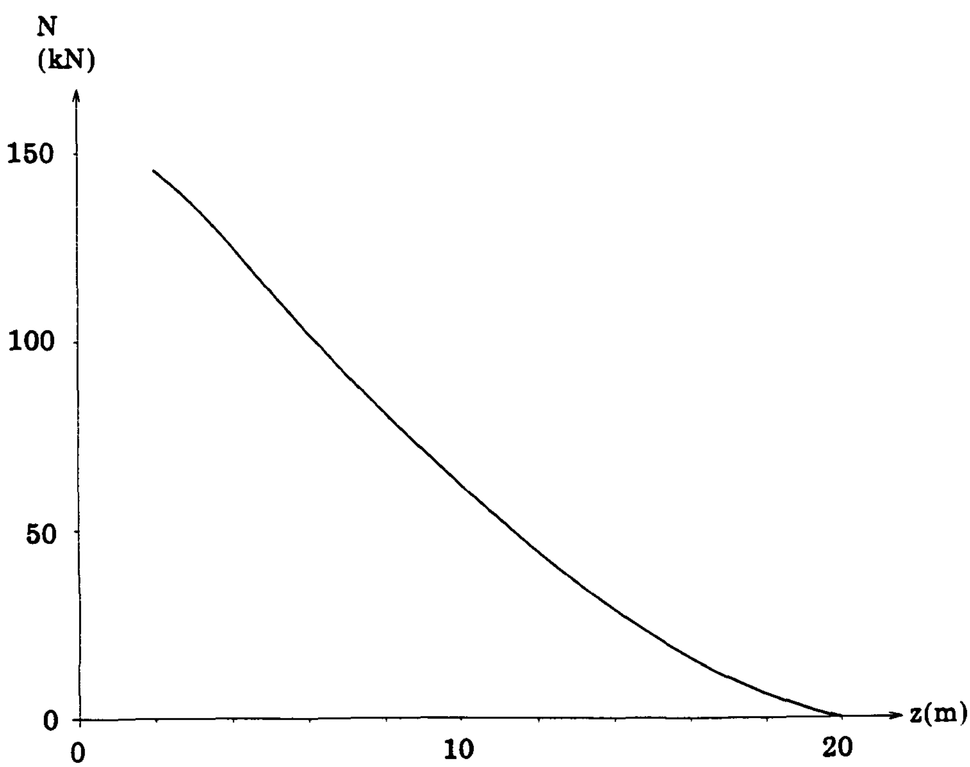


Figure 5. Normal thrust of the rotor blade caused by the centri- centrifugal force during normal operation at 3.5 rad/sec. /12

TABLE 1. ANGULAR IMPACT AND MAGNITUDES OF FORCES DURING THE 5 TYPES OF LOAD /13

Type of load	Pitch angle φ	θ	Aero-dynamic load A	Inherent weight G	Conic force K	Normal thrust N
1	0	80°	A_1	G_1	K_1	N
2	20°	80°	A_2	G_1	K_1	N
3	0	80°	A_3	G_1	$1{,}56 \cdot K_1$	$1{,}56 \cdot N$
4	20°	100°	$-0{,}5 \cdot A_3$	0	$1{,}56 \cdot K_1$	$1{,}56 \cdot N$
5	90°	80°	A_5	0	0	0

The symbols refer to Figure 6. The individual contributions of the loads are illustrated in Figures 3-5.

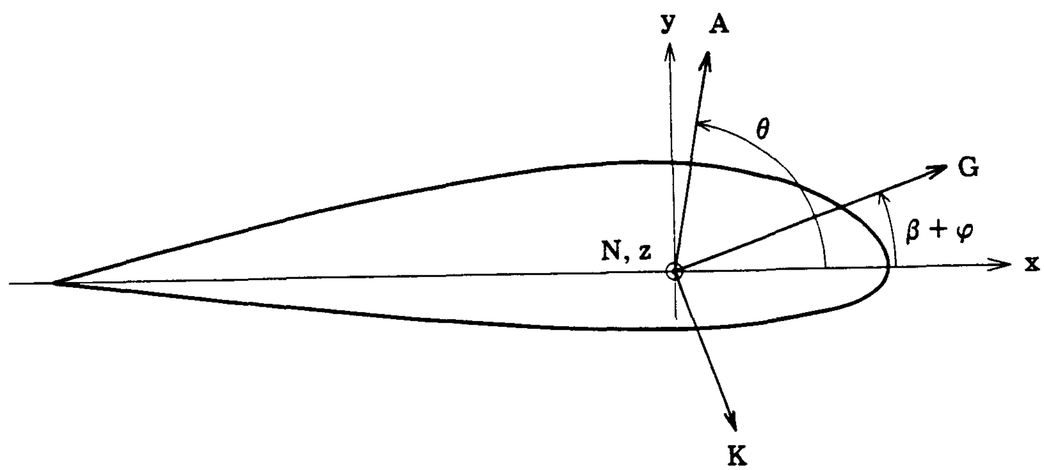


Figure 6. Blade profile with the coordinate system illustrated. Direction of the impact of the type of load is shown. The β angle is that between the profile chord and the tip chord. Other symbols same as in Table 1. /13

Maximum stress was calculated by using the cross section data obtained while applying Bernoulli's hypothesis that "plane cross sections will remain plane". The stress contribution of the normal thrust added to the bending force is at most 2% of the maximum stress calculated. The calculation of the resultant moments and of the maximum stress was done by an expansion of the computer program developed for calculating these cross section constants.

The variation of the stresses caused by the five types of load is illustrated in Figure 8. /13

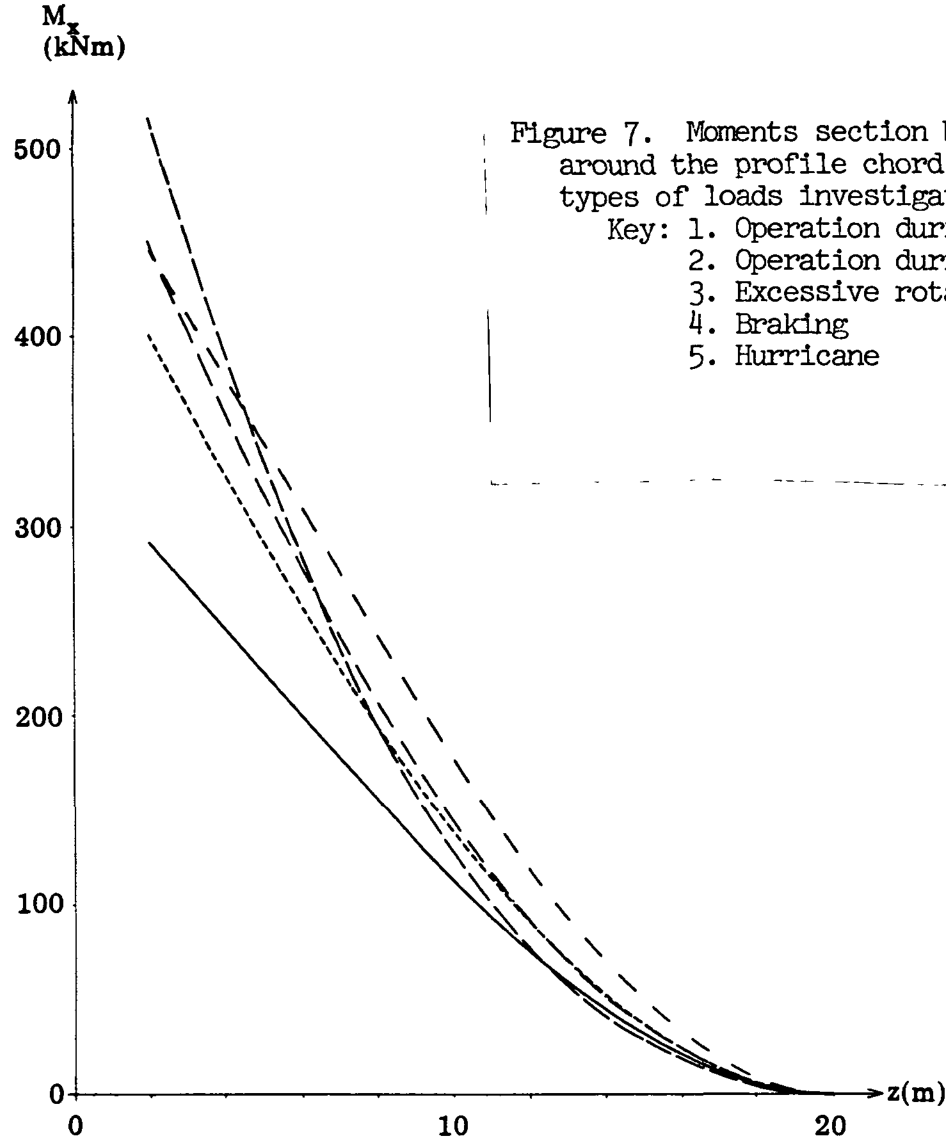


Figure 7. Moments section by section around the profile chord of the 5 types of loads investigated. /14

Key: 1. Operation during light wind
2. Operation during strong wind
3. Excessive rotation
4. Braking
5. Hurricane

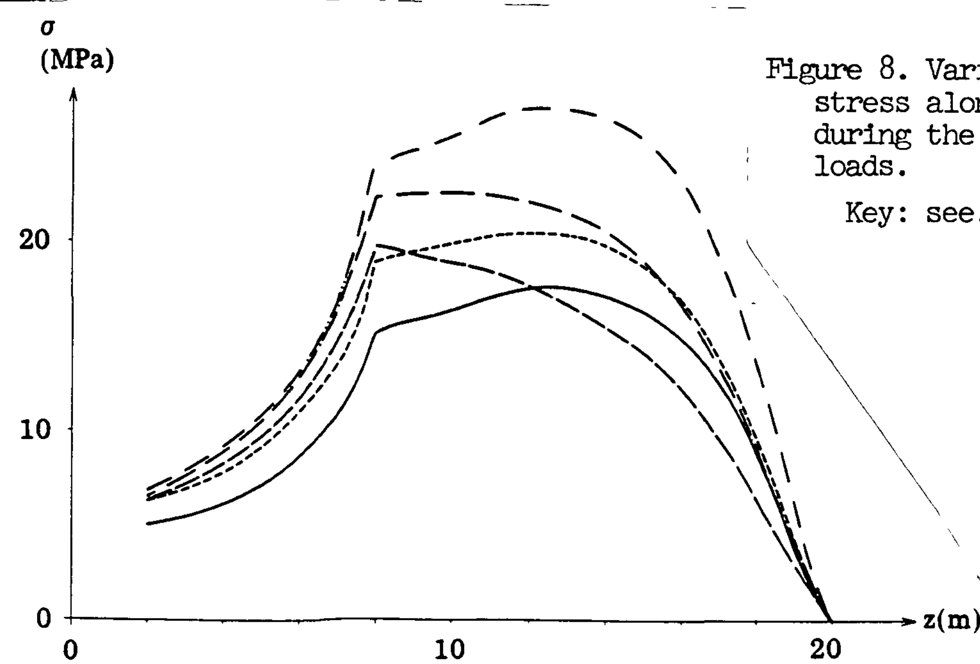


Figure 8. Variation in max. stress along the blade during the 5 types of loads. /14

Key: see. Fig.7 above.

The maximum stresses affecting the blade and illustrated in Figure 2 must be compared with the calculated bending resistance. This is obtained by reducing the typical resistance while taking humidity, duration and frequency of loading as well as the angle between the direction of the fibers and the surface into consideration; cf. [92 and [13]. /13

During load types 1 and 2 the calculated value is set at:

$$f_d = 0.4\ f_k.$$

In the case of loads 3 - 5 characterized by low frequency and short duration this value will, correspondingly, amount to:

$$f_d = 1.4 \cdot 0.4\ f_k.$$

For comparison f_d is included in Table 2. It is obvious that all the stress appearing is acceptable in every case.

TABLE 2. MAXIMUM STRESS (cf. Fig. 8) IN COMPARISON WITH CALCULATED STRENGTH DURING THE TYPES OF LOAD CALCULATED /15

Type of Load	Max. stress σ (MPa)	Calculated strength f_d (MPa)
1	17,1	20
2	20,1	20
3	26,5	28
4	22,6	28
5	19,6	28

By means of the Structural Analysis Program (SAP IV [2]) the deviations of the blade tip were calculated in parallel with and at right angle to the tip chord during type 3 of load:

$$u_x = 0.14\ \text{m}$$
$$u_y = 1.20\ \text{m}$$

In connection with the estimate of the load caused by the inherent weight of the blade investigated, an approximation of the weight of the finished blade was made. By including the weight of ribs, glue, fittings and surface /15

treatment we arrived at a total for the entire blade and its steel fittings amounting to:

$$m = 2700 \text{ kg}.$$

This calculation is given in Appendix C.

6.3 Stress Analysis: Dynamic Calculations

On the basis of the data obtained for the cross sections the Department of Fluid Mechanics (AFM) has made a dynamic analysis of the blade developed; cf. [18]. By means of a simulation program the forced response of the blade is calculated and the maximum stress moments established for a number of cases:

a. Normal operation
b. Starting/stopping during light wind
c. Starting/stopping during strong wind
d. At 25% excessive rotation
e. During a hurricane.

Case "a" shall be investigated during wind speeds of 13 m/sec when the maximum stresses occur.

Table 3 furnishes a comparison of the moments forming the basis for the preliminary dimensioning and also of the more realistic combinations of loads as calculated by the Department of Fluid Mechanics. The comparison was done for three cases:

TABLE 3. THE kNm MOMENTS OBTAINED AND USED FOR PRELIMINARY DIMENSIONING (AUC) AS WELL AS CALCULATED BY SIMULATING THE FORCED RESPONSE OF THE BLADE.

(AUC = Aalborg University Center, AFM = Dept. of Fluid Mechanics)

Position	Normal operation				Excessive rotation				Hurricane			
	AUC case 1		AFM case a		AUC case 3		AFM case d		AUC case 5		AFM case e	
z (m)	M_x	M_y	M_x	M_y	M_x	M_y	M_x	M_y	M_x	M_y	M_x	M_y
2	−291	−171	−229	−106	−445	−221	−487	−145	−516	− 91,0	−515	100
6	−200	− 88,1	−145	− 54,0	−309	−118	−320	−115	−283	− 49,8	−287	~0
10	−113	− 39,9	− 77,5	− 23,5	−177	− 55,2	−165	− 48,0	−128	− 22,6	−130	~0
14	− 44,2	− 12,7	− 27,0	− 7,0	− 69,1	− 18,1	− 57,0	− 14,0	− 39,9	− 7,0	− 41,0	~0

While "excessive rotation" and "hurricane" agree fairly well in respect to both kinds of stress, the conditions during normal operation seem to be well within the safety limits on the basis of the preliminary calculations. /18

Maximum stress has been calculated and illustrated in Table 4. These types of stress are acceptable in all the cases (cf. Hoffmeyer [3]) when consideration is given to the fact that the moments of excessive rotation and hurricane are extraordinary loads amounting to a limited number of affections during the life span of the blade.

TABLE 4. MAXIMUM STRESS IN MPa CALCULATED FOR LOADS OBTAINED WHEN SIMULATING THE FORCED RESPONSE OF THE BLADE; cf. [18].

Case of Load	Maximum stress σ			
	z = 2m	z = 6m	z = 10m	z = 14m
a	3,5	6,2	11,0	10,1
b	1,9	3,2	4,2	2,7
c	4,2	9,1	14,9	12,7
d	6,6	13,4	23,6	21,6
e	6,7	12,2	19,4	16,1

6.4 Vibration Analysis

In connection with the development of the blade profile the Structural Analysis Program , SAP IV, was used for calculating the fundamental frequencies and the type of natural vibration of the blade. The calculation model used and operating with a firmly attached blade constructed of beam elements and with concentrated mass is described in detail in [10].

On the basis of mean data for a wooden blade having the shape suggested the Department of Fluid Mechanics has made an analysis of the natural vibration (cf. Oeye [18]) corresponding to normal rotation. The data on the natural vibration are illustrated in Table 5. As is evident from the calculations the three lowest data on natural vibration lie outside the resonance in relation to the number of revolutions.

If it should become necessary the inherent frequencies of the blade can be altered by changing the mass distribution. The weight can be reduced especially at the tip of the blade, i.e. its outermost 3 to 4 meters since the stresses there are small.

TABLE 5. LOWEST NATURAL FREQUENCIES, ω, AT AN OPERATING NUMBER OF REVOLUTIONS OF Ω = 3.5 rad/sec (acc. to [18]).

Inherent frequencies	ω (rad/s)	ω/Ω
1	15,4	4,4
2	37,2	10,6
3	43,5	12,4

7. DESIGN OF ATTACHMENT

The attachment of the blade to the rotor hub has been studied in order to find a solution with the most satisfactory properties of strength. The interest centers primarily on an attachment where the shear forces are transferred through a circle of glued-in stud bolts like in the American woodes blades; cf. [5]. An evaluation of this method of attachment has been made by Riberholt [16]. A sketch of the mounting is shown in Figure 9. /17

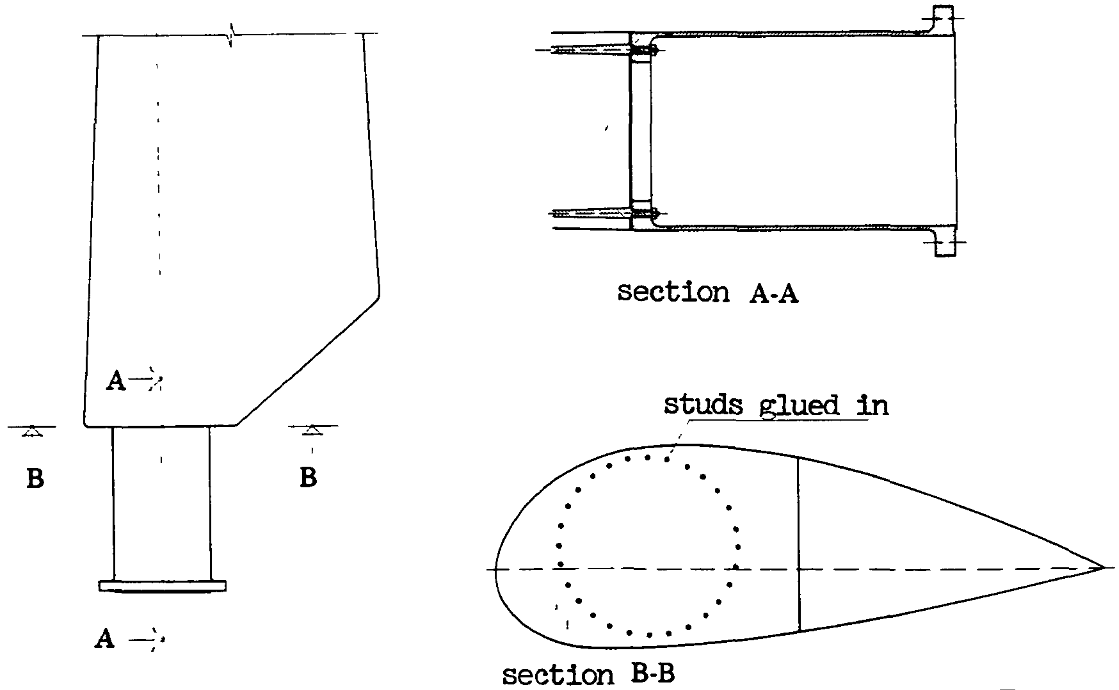


Figure 9. Details of attachment with glued-in stud bolts and a middle piece of stainless steel.

A mounting method where the supporting beam ends in a circular-cylindric tap has been contemplated. This tap of laminated wood should be wedged into a conical steel sleeve whereby the shear forces appearing would be transferred by contact pressure.

Finally the attachment could also be made by means of a split-open steel sleeve, fitting around the circular-cylindrically shaped end of the solid laminated wood part of the blade. Steel and wood could then be made to function like one piece by means of a large number of collared spikes.

On the basis of consideration given to the manufacturing process and the clearly defined static manner of operation an attachment using glued-in studs has to be recommended. A preliminary estimate of the load-bearing capacity indicates that this method of attachment permits transfer of the shear forces appearing. Because of the fatigue strength of the glued-in studs, these must be pre-stressed. The calculations are found in Appendix C. Riberholt [17] has made a dynamic calculation of this type of attachment. The computation of the steel transition has been done by Jensen [14].

It appears necessary to acquire further information based on research regarding the tensil strength of the parts used for the attachment; cf. Appendix D.

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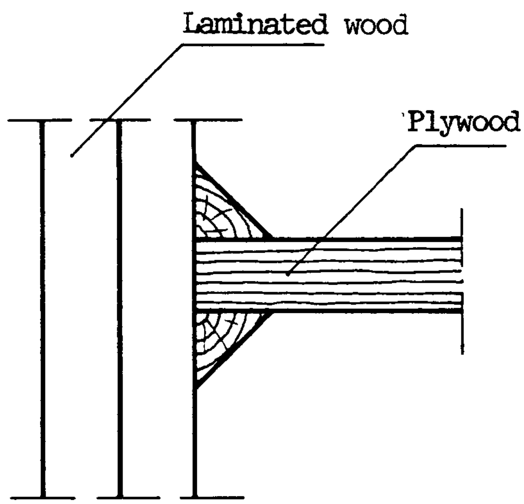
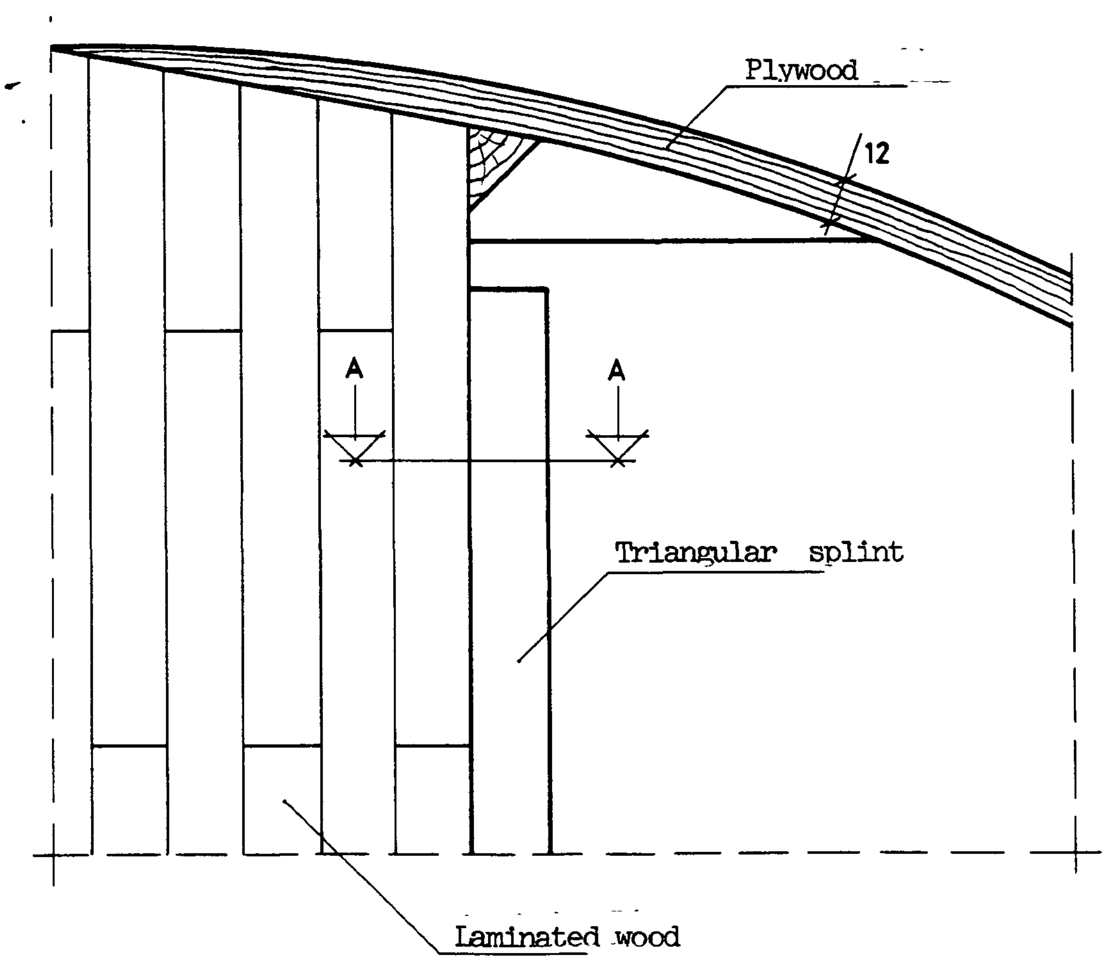
APPENDIX A

DETAILS REGARDING THE ASSEMBLING OF THE ROTOR BLADE

Figure 1. Assembling of the laminated wood and the plywood.



Section A - A



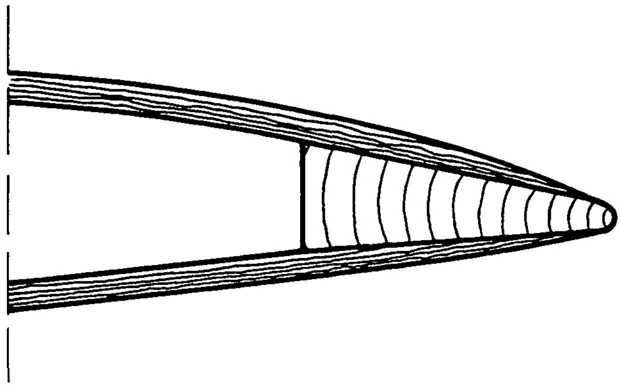
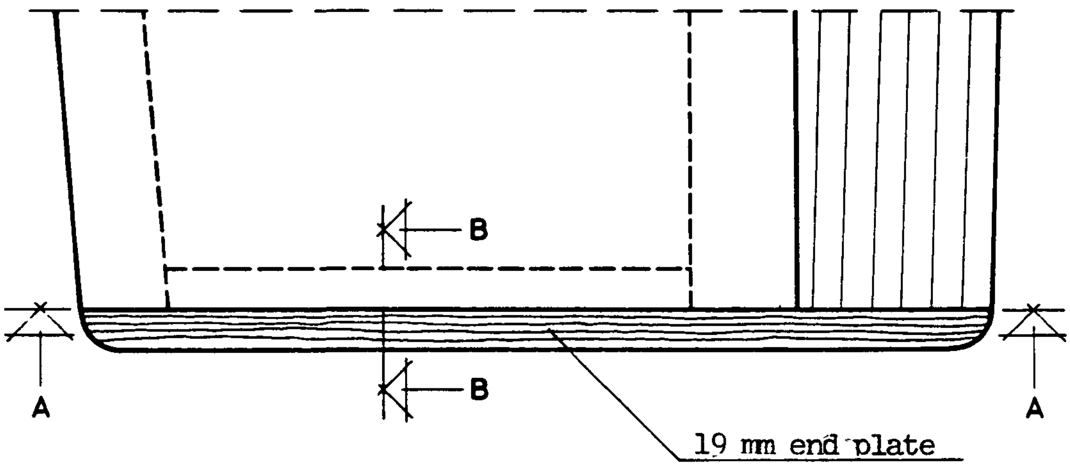
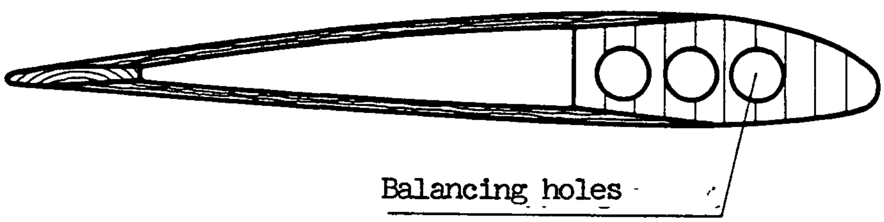
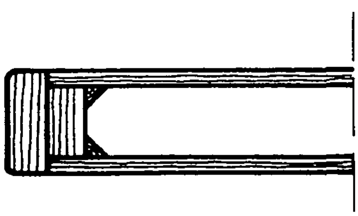


Figure 2. Mounting of the plywood panels over the trailing edge of the blade.





Section A - A



Section B - B

Figure 3. Shape of the blade tip. The principle with a grooved end plate as illustrated is also used at the root end of the blade.

APPENDIX B

Computer Program

```
      PROGRAM VIP(INPUT,OUTPUT,TAPE5=INPUT,TAPE6=OUTPUT)
C ****
C *   "VIP" UDREGNER TVAERSNITSKONSTANTER FOR NACA 4 CIFFER PROFILER
C *   MFD MASSIV D-FORMET FORSIDE OG BAGSIDE AF LUKKET TYNDFLIGET
C *   TVAERSNIT OPBYGGET I HENHOLDSVIS LIMTRAE OG KRYDSFINER.
C *   DESUDEN BEREGNES MAKSIMALE SPAENDINGER I OP TIL 6 LASTKOM-
C *   BINATIONER MED NAVIER'S FORMEL.
C ****
      DIMENSION Z(50),CL(50),BEG(50),T(50),XFC(50),YEC(50),ALG(50),
     +XP(401),YP(401),YQ(401),XPT(401),YPT(401),MBE(50),FI(6),TH(6),
     +BV(6),BG(6),BK(6),BN(6),XU(50,6),YU(50,6),SIG(50,6),BX(50,6),
     +BY(50,6)
      COMMON/A/PI,RM,RP,TP,NP,FE,FM,RUV,XM,XS1,XS2
      DATA Z1,Z2,Z3,CL1,CL3,T1,T2,T3/2.,8.,20.,2.4,.6,.80,.36,.07/
      DATA RM,RP,TH1,FE,FM,RUV/0.04,0.4,11.,0.8,1.2,450./
      DATA PI,FI,TH/3.1415926535898,0.,20.,2*0.,20.,90.,4*10.,-10.,10./
      READ(5,101)N,NP
      IF(N.GT.50.OR.NP.GT.200)GOTO 71
      READ(5,102)TP,XM1,XM2,XM3,XS11,XS12,ZS1,XS21,XS22,ZS2
      WRITE(6,200)
      WRITE(6,209)IFIX(100*XM3/CL3+.5),IFIX(1000*TP+.5)
      WRITE(6,205)
      WRITE(6,201)
      WRITE(6,203)
      WRITE(6,204)
      WRITE(6,205)
      DO 10 I=1,N
      READ(5,103)ZT,HIV,MBE(I)
      IF(MBE(I).EQ.0)GOTO 91
      READ(5,104) (BV(J),BG(J),BK(J),BN(J),J=1,6)
      BV(3)=0.25*BV(1)
      BV(5)=-BV(4)/2.
      DO 50 J=2,5
      BG(J)=BG(1)
      BK(J)=BK(1)
      BN(J)=BN(1)
      IF(J.LE.3)GOTO 50
      BK(J)=1.563*BK(J)
      BN(J)=1.563*BN(J)
   50 CONTINUE
      BG(5)=0.
   91 N2=2*NP+1
      POM=0.
      Z(I)=ZT
      CL(I)=CL1+(CL3-CL1)*(ZT-Z1)/(Z3-Z1)
      T(I)=PIP(ZT,T1,T2,T3,Z1,Z2,Z3)
      BEG(I)=TH1*(Z3-ZT)/(Z3-Z1)
      XM=PIP(ZT,XM1,XM2,XM3,Z1,Z2,Z3)
      XS1=XS11+(XS12-XS11)*(ZT-Z1)/(ZS1-Z1)
      IF(ZT.GE.ZS1)XS1=0.
      XS2=XS21+(XS22-XS21)*(ZT-Z1)/(ZS2-Z1)
      IF(ZT.GE.ZS2)XS2=0.
      CALL PROKO(XP,YP,YQ,CL(I),T(I))
      CALL TKON(XP,YP,YQ,AE,XEC(I),YEC(I),XMC,YMC,ALG(I),HIX,HIY,ZMA)
      CALL KOTRA(XP,YP,YQ,XPT,YPT,XEC(I),YEC(I),ALG(I))
      DO 60 K=2,N2
   60 POM=POM+SQRT((XP(K)-XP(K-1))**2+(YP(K)-YP(K-1))**2)
      IPO=IFIX(1000*POM)
      NAC=IFIX(RM*1.E+05+RP*1.E+03+100*T(I)/CL(I))
      WRITE(6,202)ZT,CL(I),T(I),BEG(I),XEC(I),YEC(I),ALG(I),AE,HIX,
     +HIY,HIV,XMC,YMC,ZMA,NAC
      IF(MBE(I).EQ.0)GOTO 10
      DO 20 J=1,6
      CALL MOB(FI(J),TH(J),BV(J),BG(J),BK(J),HMX,HMY,BEG(I),ALG(I),
     +BX(I,J),BY(I,J))
      SIM=0.
      DO 30 K=1,N2
      SP=ABS(BN(J)/AE+HMX*YPT(K)/HIX-HMY*XPT(K)/HIY)
      IF(SP.LT.SIM)GOTO 30
      SIM=SP
      MP=K
   30 CONTINUE
      SIG(I,J)=1.E-06*SIM
      XU(I,J)=XP(MP)
      YU(I,J)=YP(MP)
   20 CONTINUE
   10 CONTINUE
      IF(MBE(I).EQ.0)GOTO 81
      WRITE(6,200)
      WRITE(6,212)
      WRITE(6,205)
      WRITE(6,211) (J-1,J=1,6)
      WRITE(6,205)
      DO 70 I=1,N
      WRITE(6,210)Z(I),(1.E-3*BX(I,J),1.E-3*BY(I,J),J=1,6)
   70 CONTINUE
      WRITE(6,200)
      WRITE(6,206)
      WRITE(6,205)
      WRITE(6,211) (J-1,J=1,6)
      WRITE(6,205)
```

The VIP calculates the cross section constants of NACA 4 code profiles with solid D-shaped front and back edges made of a thinshelled profile of laminated wood and plywood, resp. In addition max. stress of up to 6 load combinations are calculated according to Navier's formula.

```
      DO 40 I=1,N
      WRITE(6,207)Z(I),(XU(I,J),YU(I,J),SIG(I,J),J=1,6)
   40 CONTINUE
      GOTO 81
   71 WRITE(6,208)N,NP
   81 CONTINUE
  101 FORMAT(2I5)
  102 FORMAT(6F10.3,F8.1,2F8.3,F8.1)
  103 FORMAT(F10.3,E10.2,I5)
  104 FORMAT(4E10.2)
  200 FORMAT(*1*)
  201 FORMAT(*    Z       C       T    BETA    E.-CENTER   ALFA     AREA *,
     +*       HOVEDINERTIMOMENTER   V.-STIVHED   M.-CENTER*,
     +*    MASSE PRO-*)
  202 FORMAT(*0*,F5.2,F7.3,F6.3,F5.1,2F6.3,F5.1,4E11.4,2F6.3,F7.1,I6)
  203 FORMAT(* *,27X,*X        Y*,12X,*A*,10X,*I2 *,8X,*I1 *,5X,
     +*    J        X        Y*,10X,*FIL*)
  204 FORMAT(*    M        M        M    DEG.      M       M    DEG.      M2*,
     +9X,*M4*,9X,*M4*,7X,*   M4            M       M      KG/M*)
  205 FORMAT(* ==================================================*,
     +*=======================================================*,
     +*=============*,/)
  206 FORMAT(* MAKSIMAL SPAENDING, STED (M) OG VAERDI (MPA):*,/)
  207 FORMAT(*0*,F5.2,1X,6(F8.3,F6.3,F5.1))
  208 FORMAT(*1 ### FEJL: N=*,I5,* ELLER NP=*,I5,* ER FOR STOR*)
  209 FORMAT(* TVAERSNITSDATA (MASSIV LIMTRAEDEL *,I3,* PCT. AF KORDE*,
     +* OG FINERTYKKELSE *,I3,* MM)*,/)
  210 FORMAT(*0*,F5.2,1X,6(2F9.3,1X))
  211 FORMAT(*      Z*,5X,6(*LASTTILFAELDE*,I3,3X),/)
  212 FORMAT(* MAKSIMALMOMENTER MX OG MY (KNM)*,/)
      STOP
      END
      REAL FUNCTION PIP(Z,Y1,Y2,Y3,Z1,Z2,Z3)
C ****
C *   PIP INTERPOLERER LINEAERT I FUNKTION MED ET KNAEK
C ****
      IF(Z.GE.Z1.AND.Z.LT.Z2)GOTO 51
      IF(Z.GE.Z2.AND.Z.LE.Z3)GOTO 52
      PIP=0.
      RETURN
   51 DZ=Z2-Z1
      DY=Y2-Y1
      PIP=Y1+DY*(Z-Z1)/DZ
      RETURN
   52 DZ=Z3-Z2
      DY=Y3-Y2
      PIP=Y2+DY*(Z-Z2)/DZ
      RETURN
      END
      SUBROUTINE PROKO(XP,YP,YQ,CL,T)
C ****
C *   PROKO UDREGNER KOORDINATER TIL VINGEPROFILETS
C *   UDVENDIGE OG INDVENDIGE KONTURER
C ****
      DIMENSION XP(401),YP(401),YQ(401),YC(401),YY(401),XX(401),VIN(401),
      COMMON/A/PI,RM,RP,TP,NP,FE,FM,RUV,XM
      N2=2*NP+1
      PYK=T/CL
      DO 10 J=1,NP
      XX(J)=1.-(J-1.)/NP
      X=XX(J)
      YY(J)=PYK*5*(.2969*X**.5-.126*X-.3516*X**2+.2843*X**3-.1015*X**4)
   10 CONTINUE
      YY(1)=0.
      XX(NP+1)=0.
      YY(NP+1)=0.
      DO 20 J=1,NP
      IF(XX(J)-RP)61,62,63
   61 YC(J)=RM*(2*RP*XX(J)-XX(J)**2)/RP**2
      GOTO 64
   62 YC(J)=RM
      GOTO 64
   63 YC(J)=RM*(1-2*RP+2*RP*XX(J)-XX(J)**2)/(1-RP)**2
   64 CONTINUE
   20 CONTINUE
      YC(NP+1)=0.
      DO 30 J=2,NP
   30 VIN(J)=ATAN((YC(J+1)-YC(J-1))/(XX(J+1)-XX(J-1)))
      VIN(1)=ATAN((YC(2)-YC(1))/(XX(2)-XX(1)))
      DO 40 J=1,NP
      XP(J)=CL*(XX(J)-YY(J)*SIN(VIN(J)))
      YP(J)=CL*(YC(J)+YY(J)*COS(VIN(J)))
      J2=2*NP+2-J
      XP(J2)=CL*(XX(J)+YY(J)*SIN(VIN(J)))
      YP(J2)=CL*(YC(J)-YY(J)*COS(VIN(J)))
   40 CONTINUE
      XP(NP+1)=0.
      YP(NP+1)=0.
      DO 50 J=2,NP
      J2=2*NP+2-J
      ALO=ATAN((YP(J+1)-YP(J-1))/(XP(J+1)-XP(J-1)))
      ALU=ATAN((YP(J2+1)-YP(J2-1))/(XP(J2+1)-XP(J2-1)))
      YQ(J)=YP(J)-TP/COS(ALO)
      IF(XP(J).LE.XM.OR.YQ(J).LT.0.)YQ(J)=0.
      YQ(J2)=YP(J2)+TP/COS(ALU)
      IF(XP(J2).LE.XM.OR.YQ(J2).GT.0.)YQ(J2)=0.
```

Main inertial moment, vibration rigidity, mass center, mass per -

Max. stress, position (M) and value (MPA)

Cross section data (solid lam. wood part * 13,* % of chord* and thickness of plywood

(Load case*, 13,3X)

Max. moments Mx and My

PIP interpolates linearly in function with a failure.

PROKO computes the coordinates of the external and internal contours of the blade profile.

```
   50 CONTINUE
      YQ(1)=0.
      YQ(NP+1)=0.
      YQ(2*NP+1)=0.
      RETURN
      END
      SUBROUTINE TKON(XP,YP,YQ,AE,XEC,YEC,XMC,YMC,ALG,HIX,HIY,ZMA)
C ****
C *   TKON BEREGNER TVAERSNITSKONSTANTER OG
C *   TYNGDEPUNKTSKOORDINATER FOR PROFIL
C ****
```

TKON computes the cross section constants and gravity point co-ordinates of a profile

```
      REAL IX,IY,XP(401),YP(401),YQ(401)
      COMMON/A/PI,RM,RP,TP,NP,FE,FM,RUV,XM,XS1,XS2
      XN=(XP(NP)+XP(NP+2))/4
      YN=(YP(NP)-YP(NP+2))/4
      XT=(2*XP(1)-XP(2)-XP(2*NP))/4
      YT=(YP(2)-YP(2*NP))/4
      AE=FE*XT*YT+XN*YN
      AM=FM*XT*YT+XN*YN
      SEX=0.
      SEY=FE*XT*YT*(XP(1)-XT/2)+YN*XN**2/2
      SMX=0.
      SMY=FM*XT*YT*(XP(1)-XT/2)+YN*XN**2/2
      IX=FE*XT*YT**3/12+XN*YN**3/12
      IY=FE*XT*YT*(XP(1)-XT/2)**2+FE*YT*XT**3/12+YN*XN**3/3
      CXY=0.
      DO 10 J=2,NP
      J2=2*NP+2-J
      DX=(XP(J-1)-XP(J+1))/2
      IF(XP(J+1).LE.XM.AND.XP(J).GT.XM)DX=(XP(J-1)+XP(J))/2-XM
      IF(XP(J-1).GT.XM.AND.XP(J).LE.XM)DX=XM-(XP(J)+XP(J+1))/2
      DY=YP(J)-YQ(J)
      X=XP(J)
      Y=(YP(J)+YQ(J))/2
      K=1
   72 FEX=FE*DX*DY
      FMX=FM*DX*DY
      IF(X.GE.XM)GOTO 71
      FEX=DX*DY
      FMX=FEX
   71 AE=AE+FEX
      AM=AM+FMX
      SEX=SEX+FEX*Y
      SEY=SEY+FEX*X
      SMX=SMX+FMX*Y
      SMY=SMY+FMX*X
      IX=IX+FEX*Y**2+FEX*DY**2/12
      IY=IY+FEX*X**2+FEX*DX**2/12
      CXY=CXY+FEX*X*Y
      IF(K.EQ.2)GOTO 10
      K=2
      DX=(XP(J2+1)-XP(J2-1))/2
      IF(XP(J2-1).LE.XM.AND.XP(J2).GT.XM)DX=(XP(J2+1)+XP(J2))/2-XM
      IF(XP(J2+1).GT.XM.AND.XP(J2).LE.XM)DX=XM-(XP(J2-1)+XP(J2))/2
      DY=YQ(J2)-YP(J2)
      X=XP(J2)
      Y=(YP(J2)+YQ(J2))/2
      GOTO 72
   10 CONTINUE
      K=1
      IF(XS1.LT.1.E-10)GOTO 81
      XS=XS1
   85 J=0
   82 J=J+1
      IF((XP(J)-XS)*(XP(J+1)-XS).GT.0.)GOTO 82
      YSO=YQ(J)+(YQ(J+1)-YQ(J))*(XP(J)-XS)/(XP(J)-XP(J+1))
      J=NP+1
   83 J=J+1
      IF((XP(J)-XS)*(XP(J+1)-XS).GT.0.)GOTO 83
      YSU=YQ(J)+(YQ(J+1)-YQ(J))*(XS-XP(J))/(XP(J+1)-XP(J))
      ARS=TP*(YSO-YSU)
      YS=(YSO+YSU)/2
      AE=AE+FE*ARS
      AM=AM+FM*ARS
      SEX=SEX+FE*ARS*YS
      SEY=SEY+FE*ARS*XS
      SMX=SMX+FM*ARS*YS
      SMY=SMY+FM*ARS*XS
      IX=IX+FE*(ARS*YS**2+(YSO-YSU)**3*TP/12)
      IY=IY+FE*(ARS*XS**2+(YSO-YSU)*TP**3/12)
      CXY=CXY+FE*ARS*XS*YS
   81 CONTINUE
      IF(K.EQ.2)GOTO 84
      IF(XS2.LT.1.E-10)GOTO 84
      K=2
      XS=XS2
      GOTO 85
   84 CONTINUE
      YEC=SEX/AE
      XEC=SEY/AE
      YMC=SMX/AM
      XMC=SMY/AM
      ZMA=AM*RUV
      CIX=IX-AE*YEC**2
      CIY=IY-AE*XEC**2
      CCXY=CXY-AE*XEC*YEC
```

```
      ALG=90.*ATAN(-2*CCXY/(CIX-CIY))/PI
      HIX=(CIX+CIY)/2-SQRT((CIX-CIY)**2/4+CCXY**2)
      HIY=(CIX+CIY)/2+SQRT((CIX-CIY)**2/4+CCXY**2)
      RETURN
      END
      SUBROUTINE KOTRA(XP,YP,YQ,XPT,YPT,XEC,YEC,ALG)
C ****
C *   KOTRA TRANSFORMERER KOORDINATER TIL PROFILKONTUR
C *   TIL SYSTEM I TYNGDEPKT. I HOVEDAKSERETNINGERNE
C ****
      DIMENSION XP(401),YP(401),XPT(401),YPT(401),YQ(401)
      COMMON/A/PI,RM,RP,TP,NP
      AC=COS(ALG*PI/180.)
      AS=SIN(ALG*PI/180.)
      N2=2*NP+1
      DO 10 I=1,N2
      XPT(I)=(XP(I)-XEC)*AC+(YP(I)-YEC)*AS
      YPT(I)=-(XP(I)-XEC)*AS+(YP(I)-YEC)*AC
   10 CONTINUE
      RETURN
      END
      SUBROUTINE MOB(FI,TH,BV,BG,BK,HMX,HMY,BEG,ALG,BX,BY)
C ****
C *   MOB UDREGNER MOMENTER OM DE TO HOVEDAKSER
C ****
      COMMON/A/PI
      CT=COS(TH*PI/180.)
      ST=SIN(TH*PI/180.)
      CF=COS(PI*(FI+BEG)/180.)
      SF=SIN(PI*(FI+BEG)/180.)
      CA=COS(ALG*PI/180.)
      SA=SIN(ALG*PI/180.)
      BX=BK*CF-BG*SF-BV*CT
      BY=-BK*SF-BG*CF-BV*ST
      HMX=BX*CA-BY*SA
      HMY=BX*SA+BY*CA
      RETURN
      END
```

KOTRA transforms coordinates into profile contours for the system in the gravity point in direction of the main axes.

MOB calculates moments around the two main axes.

APPENDIX C.

RESULTS OF THE CALCULATIONS /C1

Data on Cross Sections

The cross section constants used are shown in Table C. These constants are calculated on the basis of the computer program illustrated in Appendix B. The torsion inertia moment is, however, only approximated as a mean of values obtained for a solid and for a hollow blade profile respectively. The formulas used are:

$$J = \frac{4A^2 \cdot t}{s} \qquad \text{(for a hollow profile)}$$

$$J = \frac{4I_x}{1 + 16\,\dfrac{I_x}{A \cdot c^2}} \qquad \text{(for a solid profile)}$$

where

A = area
c = the chord length of the profile
I_x = the bending inertial moment around the chord
s = the length of the curve along the center line of a thin-shelled profile
t = the thickness of the shell

TABLE C1. CROSS SECTION DATA USED.

Cross section data (solid laminated wood: 33% of chord, plywood thickness 12 mm.

Z M	C M	T M	BETA[1] DEG.	E-CENTER[2] X M	Y M	ALFA[3] DEG.	AREAL A M2	HOVEDINERTIMOMENTER (main inert. mom.) I2 M4	I1 M4	V.I.MOMENT J M4	M-CENTER[2] X M	Y M	MASS KG/M
2.00	2.400	.800	11.0	.597	.080	1.8	.7164E+00	.3254E-01	.8924E-01	.1000E+00	.619	.079	328.6
3.00	2.300	.727	10.4	.566	.075	1.7	.6090E+00	.2287E-01	.7187E-01	.7330E-01	.589	.075	280.1
4.00	2.200	.653	9.8	.536	.070	1.5	.5102E+00	.1552E-01	.5749E-01	.5220E-01	.561	.070	235.5
5.00	2.100	.580	9.2	.507	.066	1.4	.4202E+00	.1011E-01	.4572E-01	.3580E-01	.535	.065	194.8
6.00	2.000	.507	8.6	.480	.061	1.2	.3388E+00	.6251E-02	.3617E-01	.2340E-01	.511	.061	158.0
7.00	1.900	.433	7.9	.456	.056	1.0	.2663E+00	.3621E-02	.2843E-01	.1440E-01	.491	.056	125.2
8.00	1.800	.360	7.3	.435	.051	.8	.2025E+00	.1925E-02	.2212E-01	.8430E-02	.476	.051	96.3
9.00	1.700	.336	6.7	.416	.049	.8	.1798E+00	.1494E-02	.1819E-01	.6570E-02	.457	.049	85.8
10.00	1.600	.312	6.1	.398	.046	.7	.1584E+00	.1139E-02	.1478E-01	.5030E-02	.437	.046	75.9
11.00	1.500	.288	5.5	.379	.043	.7	.1384E+00	.8510E-03	.1183E-01	.3770E-02	.418	.043	66.5
12.00	1.400	.263	4.9	.360	.040	.6	.1197E+00	.6210E-03	.9315E-02	.2770E-02	.399	.040	57.8
13.00	1.300	.239	4.3	.341	.037	.6	.1024E+00	.4407E-03	.7184E-02	.1980E-02	.379	.037	49.7
14.00	1.200	.215	3.7	.322	.034	.5	.8634E-01	.3025E-03	.5413E-02	.1360E-02	.359	.034	42.2
15.00	1.100	.191	3.1	.303	.031	.5	.7164E-01	.1994E-03	.3962E-02	.9020E-03	.339	.031	35.3
16.00	1.000	.167	2.4	.283	.028	.5	.5824E-01	.1248E-03	.2792E-02	.5720E-03	.318	.029	28.9
17.00	.900	.143	1.8	.263	.026	.5	.4613E-01	.7303E-04	.1875E-02	.3380E-03	.296	.026	23.1
18.00	.800	.118	1.2	.242	.023	.6	.3527E-01	.3899E-04	.1176E-02	.1830E-03	.272	.023	17.9
19.00	.700	.094	.6	.219	.021	.8	.2556E-01	.1817E-04	.6695E-03	.8750E-04	.246	.021	13.2
20.00	.600	.070	0.0	.192	.019	1.2	.1681E-01	.6694E-05	.3330E-03	.3570E-04	.216	.019	8.7

1) Angle chord/tip chord; 2) Elasticity center, point of gravity and mass center. Distance from profile nose - x - and chord - y - is given.
3) Angle between 2nd main axis and profile chord.

Calculation of Weight

In connection with the calculation of the data concerning the cross sections /C2 we estimated the weight per length unit of sections 1 m apart along the entire blade; cf. Table C1. On the basis of these figures the weight of the entire blade can be estimated as follows:

Weight of wooden blade	1750 kg
Weight of ribs	50 kg
Weight of glue and splints	170 kg
Weight of surface treatment	130 kg
Weight of fittings (nails, and studs)	100 kg
Weight of steel sleeve at attachment	500 kg
In total	2700 kg

All weights are approximate.

Preliminary Calculations Regarding the Attachment

The maximum shear forces between the blade and the hub at 2 m from the center of rotation are shown in Table C2.

TABLE C2. SHEAR FORCES AT THE ROOT OF THE BLADE DURING MAXIMUM POSSIBLE WIND LOAD; THE x-AXIS IS PARALLEL TO THE CHORD OF THE BLADE.

Type of Load	Normal thrust	Dislocation forces (kN)			Moments (kNm)		
	(kN)	Q_x	Q_y	$\lvert Q_{res} \rvert$	M_x	M_y	$\lvert M_{res} \rvert$
1	145	25,8	22,0	33,9	−291	171	338
2	145	30,3	39,2	49,5	−403	209	454
3	227	30,7	32,6	44,8	−445	221	497
4	227	16,9	−46,7	50,0	452	144	474
5	−16,4	11,8	67,2	68.2	−516	91	524

The most hazardous load on the assembly occurs during type 3 of load.

The connection between the parts consists of a circle of 24 conic glued-in studs; cf. Figure 9. The blade is attached to the existing flange of the hub by means of a middle piece consisting of a steel sleeve with flanges at both ends. By using maximum strength steel with milled thread and a glued-in length of 30 x diam. of studs it appears that it will be possible to achieve the calculated strength of the studs at an axial load of 120 kN and a shear stress of 15 kN.

If the circle of studs having a diameter of $d = 680$ mm is made of 28 pieces of studs, the distance between the studs will amount to:

$$a = \frac{\pi \cdot 680}{28} = 76 \text{ mm.}$$

The maximum load on the studs in relation to normal thrust and moments will be:

$$P_0 = \frac{2 \cdot 497}{28 \cdot 0.340} + \frac{227}{28} = 113 \text{ kN.}$$

In relation to the dislocation force it will be: /C3

$$P_{90} = \frac{45}{28} = 1.6 \text{ kN.}$$

Since

$$\left(\frac{113}{120}\right)^2 + \left(\frac{1.6}{15}\right)^2 = 0.90 < 1$$

it seems the mounting will be able to transfer the stress applied.

APPENDIX D.

EXPERIMENTAL RESEARCH /D1

A condition for the total evaluation of the wooden blade project and for the final decision on the shape of the blade is that certain phenomena observed must be verified by experiments. In this connection it is necessary to distinguish between the experiments which offer an opportunity for building a set of blade prototypes and the more extensive and general tests necessary before serial production of wooden blades can be started.

Below a short description is given of the experiments necessary before a set of prototypes can be constructed.

Tests on Conic Studs to be Glued in

Static and dynamic tests suggested by Riberholt (cf. Supplement C) will be made on the studs to be glued in.

Tensile strength tests on two differently long glued-in portions will be made in order to establish short-term fatigue. When the length of the glued-in portion has been decided, fatigue limit tests will be conducted in the form of three series of tests where the load varies between 0, 40, 50 and 60% of that of the short-term resistance.

Test of Attachment

A root end section complete with fittings and circle of glued-in studs will be constructed. This assembly will be loaded and the stress on the studs measured. The short-term fatigue limit of the attachment shall also be measured.

Material Tests on Laminated Wood

In order to test bending resistance and rigidity of the suggested construction of the beam of laminated wood, tests will be made both with cross-glued as well as conventionally laminated wood. Special emphasis will be placed on establishing the effect of the fact that the wood fibers are angled against the surrounding surfaces.

At least three series of tests should be made on beams with constant diameter. During the first series of tests all the laminating boards will have fibers in parallel with the sides of the beam, while during the second series the fibers will form a 4^{o} angle with the upper surface of the beam corresponding to a slanted trim of the blade and a maximum permissible fiber angle of the boards. The third test series will consist of cross-glued beams. The fibers of every second layer of laminating boards form an 8^{o} angle with the upper surface of the beam. The fibers of all the other layers will be parallel with this surface.

All the laminating boards used for the test beams must be edge-glued. Before assembling, the rigidity of the individual boards must be tested for comparison with the rigidity of the finished beam.

Beams will be selected from the first test series for preliminary vibration tests. Both natural and forced vibration will be used for establishing the natural frequencies and the nodal damping conditions of the beams. The importance of the moisture content and the effect of the vibration rate will also be established.

A few beams will be selected from the first test series as well in order to be tested in respect to their bending fatigue limits.

Tests on Blade Sections

A section of blade with 1.5 m chord length and 260 mm diameter will be built according to the dimensions of the blade where the maximum stress occurs.

The blade section will be subjected to natural and forced vibrations in order to establish its natural frequencies and their damping. Therefore the blade section will be subjected to such bending stress that data on strength and rigidity obtainable in practise can be estimated. The production of such a test piece will provide the manufacturer with an opportunity for testing the production technology in respect to both the gluing procedures and the trimming of the NACA profile. /D2

Instrumentation of Test Blades

When the test blades have been constructed, their natural vibration frequencies can be determined. The blades will be provided with instrumentation for measuring the bending stress at the root end and the moments at the most severely affected cross sections as well as the moisture content during operation.

PRODUCTION OF WOODEN BLADES FOR WIND TURBINE B AT NIBE

Preben Hoffmeyer

WOOD TECHNOLOGY /I

Technical Institute

PRODUCTION OF WOODEN BLADES FOR WIND TURBINE B AT NIBE

CHOICE OF MATERIAL

by

Preben Hoffmeyer

Translation of "Fremstilling af traevinger til Nibe moelle B. Materialevalg." Teknologisk Institute, January 18, 1982, pho/bs-2213454/60 , 8 p. + Appendix.

Technical Institute

January 1981

LIST OF CONTENTS

Summary

This report contains an evaluation of suitable qualities of construction wood and construction plywood for use in the manufacture of wooden rotor blades for the wind turbine B at Nibe.

1. Load-bearing Beam made of Laminated Wood

1.1 Dimensions

The load-bearing beam made of laminated wood will be 20.0 m long and attached over a distance of 2.0 m from the centre of rotation. The cross section of the beam will be gradually tapered in two stages with the transition point about 8 m from the center of rotation.

The tapering of the beam amounts to 1:30 for the section between 2.0 and 8.0 meters which is most subjected to stress and to 1:80 for the section between 8.0 ànd 20.0 m. The intended position of the laminating boards means that there will be an "inherent" slanting of the fibers corresponding to the tapering.

1.2 Stresses

Preliminary calculations indicate that maximum bending stress during normal operation (case 2) will increase rapidly to ca. 17.8 MPa at the 8 m point and thereafter grow more slowly until reaching a maximum of ca. 18.6 MPa around the 12 m point.

The worst case of extraordinary stress (i.e., excessive speed) results in maximum bending stresses of ca. 22.4 MPa and 24.8 MPa at the 8 m and 12 m points, respectively.

The determining load moment is the constantly pulsating bending stress at a maximum of ca. 18 MPa since we must therefore take a reduction in strength as a consequence of long-term stress into consideration.

1.3 The Strength of the Laminating Boards

The typical resistance to short-term fatigue, f_m, at a moisture content of 12% can be calculated according to the formula:

$$f_k = f_m \cdot \gamma_{moisture} \cdot \gamma_{mat}/\gamma_{lam}/\gamma_{fiber}/\gamma_{time},$$

where f_m is the estimated bending resistance and $\gamma_{moisture}$ is a factor indicating that moist wood is weaker than dry wood. The average moisture content of the laminated wooden beam must correspond to the moisture content of surface wood for use out-of-doors. This will of course depend on the choice of surface treatment and the ventilation of the blade. The average moisture content of the beam will hereafter be considered to be ca. 18%. The bending resistance corresponding to a 12% moisture content can be obtained by calculating with an increase in resistance by ca. 3% per 1% reduction in moisture which would result in $\gamma_{moisture} = 1.18$.

The expression γ_{mat} stands for the partial coefficient of the material. /2
It is usually set at 1.3. The determination of the coefficient in this case can actually be done only after an analysis of the total safety problem. In order to be able to already at this point in time make a statement regarding the quality of wood most suitable for the load-bearing beam, it is necessary to make an estimate. Considering the intended extreme care to be taken when selecting the laminating boards (among others by automatic sorting according to tensile strength) and the careful manufacturing of the beam, a factor of $\gamma_{mat} = 1.2$ can be considered reasonable.

It is normal to consider the strength of the laminated wooden beam as greater than the strength of the individual boards of which it is made. The laminating factor, γ_{lam}, is especially significant when low quality wood is laminated. Where - like in this case - it is a question of boards with extremely few imperfections the laminating effect is usually more moderate. It is a special property of this supporting beam that the lamination runs perpendicularly in relation to the direction of the stress. This means that many boards are simultaneously within the zone most affected. At the same time the boards are thinner than ordinarily. On the basis of these facts the laminating factor should be estimated to $\gamma_{lam} = 1.2$.

The expression γ_{fiber} concerns the reduction in strength depending on the slanting of the fibers. Using an expected brutto fiber slant (trimming + the natural slant of the fibers) of 1:15, the γ_{fiber} can be estimated to = 0.9.

The long-term factor, γ_{time}, concerns bend-stressed wood. This factor is valid for high quality wood when set at 0.6. Because of complicating conditions such as, e.g., the slanting of the fibers, vibration effects, etc. (cf. the Appendix) a time factor of $\gamma_{time} = 0.5$ must be considered most reasonable. Here no consideration will be given to the fact that the load is not always maximal (18 MPa) which bestows a moment of safety on γ_{time}. How much this amounts to must be investigated during the project described in the Appendix.

Thus, f_k, can be estimated as follows:

$$f_k = 18 \cdot 1.18 \cdot 1.2/1.2/0.9/0.5 \approx 47 \text{ MPa}$$

1.4 Types of Wood and Sorting

The requirement demanding a typical bending strength of 47 MPa can be met by softwood of Scandinavian origin. The planned dimensions of the wood, i.e. 25 x 125 mm or broader, can be cut in the form of side boards (which have few knots). Widths up to 125 mm and more can often be cut from logs of large dimensions (i.e. with insignificantly slanting fibers).

It has been pointed out in a Swedish report [1] that the yield of T50 during automatic sorting of V-quality spruce logs amounts to a few percent in respect to 50 mm boards but increases to about 10% in the case of 35 mm boards. It can be expected that a still better yield can be obtained for 25 mm boards.

We therefore recommend the use of 25 mm automatically sorted Norway spruce.[1] /3
This dimension is the smallest which can be sorted by the only existing Danish automatic stress sorting machine.

This dimension seems to be used only infrequently at laminating wood factories and it may be necessary to order the raw material for the test blade separately. Normally Norway spruce grown in Sweden is used for manufacturing laminated wood. It should, however, be mentioned that Norway spruce from certain Danish localities in Jutland are fully comparable to the spruces grown in Sweden.

If concern regarding the strength of the splices (cf. 1.5) makes it necessary to use extra thin boards, these can easily be cut from 38 mm automatically sorted boards since these are the standard size used for the manufacture of laminated wood. The advantage is that certain Danish factories making laminated wood traditionally used Swedish Norway spruce of a quality which yields a high output during automatic sorting corresponding to 47 MPa.

The price of Norway spruce of the quality mentioned varies between 1400 and 1600 Danish Crowns/m^3. Automatic sorting of 10 m^3 for a test blade could be done for a cost price of ca. 140 D. Cr./m^3.

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[1] Translator's note: *Picea Abies*.

The automatic sorting according to strength must be supplemented by visual inspection, especially concerning storm damage (internal breaks) and the slanting of the fibers along the broad side of the board. Signs of storm damage must be checked for on the boards when they have been planed.

The greatest problems concerning strength occur around the 8 m point where the bending stress is almost at maximum and the natural slant of the wood fibers is a constant of 1:30. Be requiring a maximum "brutto fiber slant" of 1:15 a limitation of the natural slant of the fibers to 1:30 (when measured on the broad side) becomes necessary.

When the load-bearing beam is being trimmed a smoothly proceeding tapering at the 8 m point is adamantly required.

1.5 Splices

While it is hardly any problem to obtain wood with an f_k = 47 MPa, there may arise a problem making the splices strong enough. It is almost impossible to avoid splicing within the zone of maximum stress since normal operation (case 2) results in high tensions over a distance of 8 m.

An extraordinary high quality of splicing must be demanded. Research must prove that the tensile strength of these splices is adequate. The question may even arise whether "test loading" of the segments belonging to the zone must subject to stress is not necessary.

Since "perpendicular lamination" is used for the load-bearing beam the individual boards must be spliced so that "fingers" are visible only on the narrow side of the boards; this prevents any unfortunate lateral stress on the glue seams of the fingers. Diagonal splicing should be considered. The placement of "fingering splices" around the 8 m point should be avoided. /4

If it should prove too difficult to produce adequately strong splices, the problem can be solved by using "super quality boards" (i.e. thin boards where the imperfections of each individual board is negligible in relation to the total cross section). The fact should also be considered that not all the boards within the cross section of the beam are heavily loaded so that a differentiated arrangement of wood qualities might pay for itself.

1.6 North American Softwoods

The use of North American softwoods (Sitka Spruce, Douglas Fir /sometimes called Oregon Pine/ or Western Hemlock[1]) has been contemplated. These woods come from such large logs that the requirement for straight fibers and freedom of knots should not present any major problems during the sorting.

It is, however, difficult to obtain boards longer than 10 m because of which it will be necessary to splice the American wood as well. Since at the same time the price is almost three times that of Scandinavian softwood, only very special conditions will make the use of North American wood necessary, such as:

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[1] Translator's note: *Picea Sitkensis, Pseudotsuga Menziesii, Tsuga heterophylla, respectively.*

1. if it should prove difficult to find Norway spruce with fibers slanting less then 1:30, or

2. if it should be absolutely necessary to purchase boards longer than 10 m.

In respect to the three kinds of wood which could come into question the following can be briefly stated:

Sitka spruce has for a long time been used for the manufacture of airplanes and ships because of its excellent elasticity and its straight fibers. It is well known that the English company Permali Gloucester Ltd. uses Sitka spruce for the manufacture of large wind tunnel propellers [2, 3]. Both Sitka spruce and Douglas fir are recommended by this company for the production of large rotor blades [4]. Sitka spruce is still imported to Denmark (by Odense Lumber Yard) in small amounts for use within the shipbuilding branch, especially for making masts. The common dimensions are 2" - 2 1/2" x 6" - 9" in lengths up to 20'. The delivery time for other dimensions is 3 to 5 months. The price is ca. 6000 D. Cr./m^3 (including kiln drying).

Douglas fir (also called Oregon pine) is imported in large quantities /5
to Denmark. The quality which can come into question here is that of so-called "mirror finished" deck planking which is used within the furniture industry as well. It is the kind of wood used succesfully for an American project concerning large wooden rotor blades. The typical dimensions imported are 2 3/4" x 5 1/4" in lengths up to 20'. Douglas fir of the quality discussed costs ca. 6500 D. Cr./m^3.

Western hemlock is also imported to Denmark in large quantities which are, however, hardly adequate for further cutting into boards for laminated wood since the wood is frequently full of both visible and especially of invisible knots. However, if Western hemlock of the correct quality and dimensions could be imported, this type of wood may prove equally useful as Sitka spruce and Douglas fir.

A study of the mechanical properties of Sitka spruce, Douglas fir, Western hemlock and forest pine (*Pinus silvestris*), all of prime quality, is published in [6].

1.7 Elasticity Module, Moisture and Seasonal Temparature Dependencies.

The mean elasticity coefficient for automatically sorted Norway spruce with a typical tensil strength of 47 MPa will at 12% moisture content be about 15000 MPa. The variation coefficient may amount to ca. 15%.

If because of the danger of resonance problems it is desirable to delimit the elasticity module downward as well as upward, this can be achieved without difficulty. It seems that the variation coefficient of the elasticity module can be lowered to $\pm$10% for individual boards. The inequality of the elasticity module (as well as the density) of the load-bearing beam seems to be neglible in such a case.

The mean elesticity coefficient at the 18% moisture content mentioned should be ca. 13500 MPa since the elasticity module changes in relation to every 1.5 to 2% change in moisture content.

When estimating the rigidity of the beam it is necessary to take into consideration that the changes in moisture content also implies altered dimensions.

The seasonally related variations in moisture content of the laminated wooden beam may be reduced due to the large dimensions of the beam and due to the fact that the surface treatment slows down the circulation of the moisture. Although seasonal variations out-of-doors usually result in a moisture content of the wood varying between 20 - 24% during winter and occasionally as little as 12% during summer, the variation in moisture content in the case discussed seems limited to the interval 16 - 20%. However, the variations could be larger at the tip end and smaller as the base of the blade.

In correlation with the variations in moisture content, the elasticity /6 module may vary between ca. 14000 MPa (during summer) and ca. 13000 MPa (during winter).

Long-term stress on a wooden construction may result in a decreasing elasticity module. It cannot be judged in advance by how much the vibration stress will reduce the elasticity module of the beam. An effort to explain this must await a calculation of the consequences of vibration.

1.8 Type of wood for the Attachment of the Blade

The planned attachment of the rotor blade by means of glued-in studs [7] or some other form of steel/wood glued connection may prove to be limited by, e.g., the dislocation stress of the wood and/or its shear stress under pressure.

It will in this connection be possible to meet the requirements on greater strength by locally using laminating boards of a wood stronger than that of Norway spruce. This has been done, e.g. in the case reported in [2]. A suitable wood type could be Honduras mahogany[1] which takes excellently to gluing and which could resist stresses up to 50% larger than Norway spruce is able to do.

2. PLYWOOD

From a study of the literature, especially that concerning airplane manufacture, it is evident that birch plywood has always been the preferred type of plywood. At present many kinds of useful types are produced especially for shipbuilding purposes but in respect to the price we recommend the use of Finnish birch[2] plywood for the wooden rotor blade.

[1] Translator's notes: *Swietenia macrophylla*, [2] *Betula pendula*.

The choice is made easier due to the fact that Finnish birch plywood is manufactured with a number of different surface treatments, among others, a suitable glassfiber reinforced polyester coating on both sides (e.g., the brand Schauman's "Wisacon"). In addition, the plywood can be delivered already treated against rot.

The Finnish birch plywood can be delivered in formats as large as 3 x 20 m, depending on brand. A smaller format can be selected for easier transport and handling and the assembly can be made simpler by precut diagonal (1:10 splicing edges through the entire thickness of the panel. The normal thicknesses are 9 or 12 mm (corresponding to 7 or 9 plys, respectively).

The thickness of the surface coating can be selected within a range of ca. 800 to 2400 g/m^2, corresponding to an increase in thickness of the panel from 0.5 to 2.0 mm.

The price of 9 or 12 mm plywood including surface coating as mentioned ranges between 200 and 250 D. Cr/m^2.

3. SURFACE TREATMENT AND IMPREGNATION /7

It is assumed that the hollow space within the rotor blade shall be well ventilated.

The external surface of the rotor blade can be surface treated by applying a strong glassfiber reinforcement and a finish consisting of a painted coat of acrylic-enhanced polyurethane.

The plywood should be delivered rot-proof but an additional surface coat (of acrylic-enhanced polyurethane) may be considered.

4. ADDENDUM

On the basis of the most recently made estimates of the stresses on the blade (Appendix B), such can be expected which make it possible to use wood with a calculated bending resistance at a level 30 to 40% lower than originally supposed in this report.

The project committee has, however, decided to include this excessive resistance as an additional contribution to the safety limits. After verification of the calculations by full scale experiments with the load-bearing beam it can, if necessary, be decided to reduce the dimensions of the beam or permit the use of wood of a lesser quality than suggested here.

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The evaluation of the correlation between typical short-term fatigue and fatigue limit is marred by uncertainty in that

- the maximal load (18 MPa) is not constantly active;
- the correlation between bending resistance and the slanting of the fibers (the 0.9 factor) is valid for short-term effects only. On the other hand, time is considered more important for the shear resistance (time factor = 0.4) than for the bending force in parallel with the fibers (time factor = 0.6). This may imply that the importance of the fiber slant increases by time and that the 0.9 factor is too large;
- the normal time factor (0.6) for bending high quality wood in paralell with the fibers is based on static tests. There is sparse information regarding dynamic tests.

  Kollmann [10] and the Wood Handbook [11] indicate that the time factor for high frequency bending tests may be as low as 0.30. Here it is, however, a question of twisting and bending tests whereby alternately tension and pressure stresses are active. The low time factor can therefore be explained by the fact that the eventual failure has been prepared for by cracking due to buckling under pressure. For a unilaterally pulsating load the time factor is said to vary between 0.60 for wood free of imperfections and 0.30 for wood with small knots and a fiber slant of 1:12.

  Bach [8] has indicated that the time factor depends on the frequency in the case of wood free of imperfections and subjected to a low frequency pulsating pressure. Research demonstrates that the time factor will become less important, the lower the frequency applied is.

  During a correspondence with W. V. Roth we have learnt that preliminary results of fatigue tests on construction wood [9] indicate a time factor of 0.5 to 0.6 for a pulsating bending stress.

On the basis of what is mentioned above and the incomplete information available, we recommend that a project be started for research on: "The importance of pulsating bending stress for the rigidity and life span of construction wood with special attention to the importance of lateral tensile strength of boards as well as of splices". This project should lead to an elucidation of the most likely history of stress of a rotor blade intended for the Nibe wind turbine as well as information on how this stress should be included into the estimates on its life span.

REPORT III

CALCULATIONS CONCERNING STEEL SLEEVES FOR ATTACHING
THE NEW WOODEN ROTOR BLADE

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CALCULATIONS CONCERNING STEEL SLEEVES FOR ATTACHING THE NEW WOODEN ROTOR BLADE

by

Ejgil Jensen

Translation of "Beregning af staalroer til vingebefaestigelse paa nye traevinger", DEFU, EJ/bin, March 16, 1982

DEFU Research Institute of the Danish Electric Power Companies
Electric Technology - Computer Programming - Energy Problems.

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1. INTRODUCTION /1

The sleeve is planned to function as the middle piece between the hub and the wooden rotor blade. Two alternative designs are shown in Figures 1 and 2. The attachment of the hub begins at R = 0.7 m and the wooden blade takes over at R - 2.0 m. The attachment to the hub will be accomplished by 36 stud bolts, Ø 33, arranged in a circle with a diameter d_1 = 910 mm. It is planned that another 25 studs shall be glued into the wooden butt in the form of a circle with d_2 = 650 mm. The alternative in Figure 2 requires, however, two circles of which the outer must not be wider than the d_2 = 650 mm mentioned above because of the distance to the free edge.

The stress on the studs is supposed to the transferred to the sleeve via the flanges illustrated. The flanges must be adequately stiff in order to be able to stand up to the great bending stresses on the studs due to the eccentricity forces. The second alternative (Figure 2) reduces considerably (eliminates?) these bending stresses on the glued-in bolts.

However, during preliminary calculations concerning alternative 2, it proved to be unrealistic. The arrangement would make the construction more expensive and it would be almost impossible to mill such a sleeve because of its dimensions. For the sleeve according to alternative 1 a piece of material 16 mm thick will be required whereas - for safety reasons - a thickness of 28 mm would be required for the piece to make alternative 2. This is unrealistic. Furthermore it seems that the flange of alternative 1 can be given adequate rigidity in order to fulfill the requirements of small-size angular torsion. Therefore only alternative 1 will be considered below.

2. PREREQUISITES /4

Type of steel:	St 37.3	(DIN 17100)
Certificate:	3.1 B	(DIN 50049)
Quality of welding:	class C	(SVC 739022)

Calculations made according to Danish Standards: DS 412 And DS 412.2.

Construction class: 3
Material control: 2

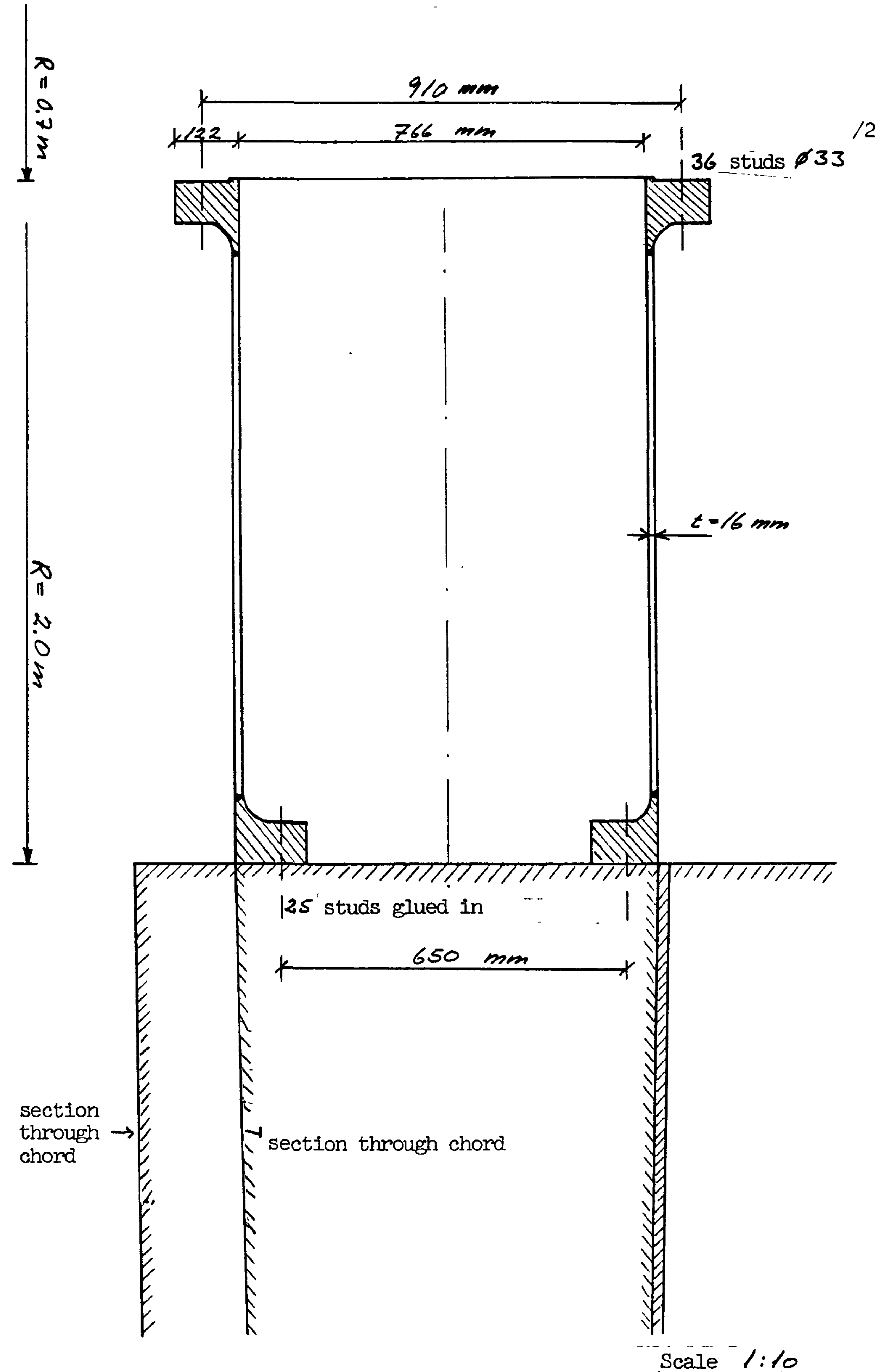


Figure 1. Alternative 1.

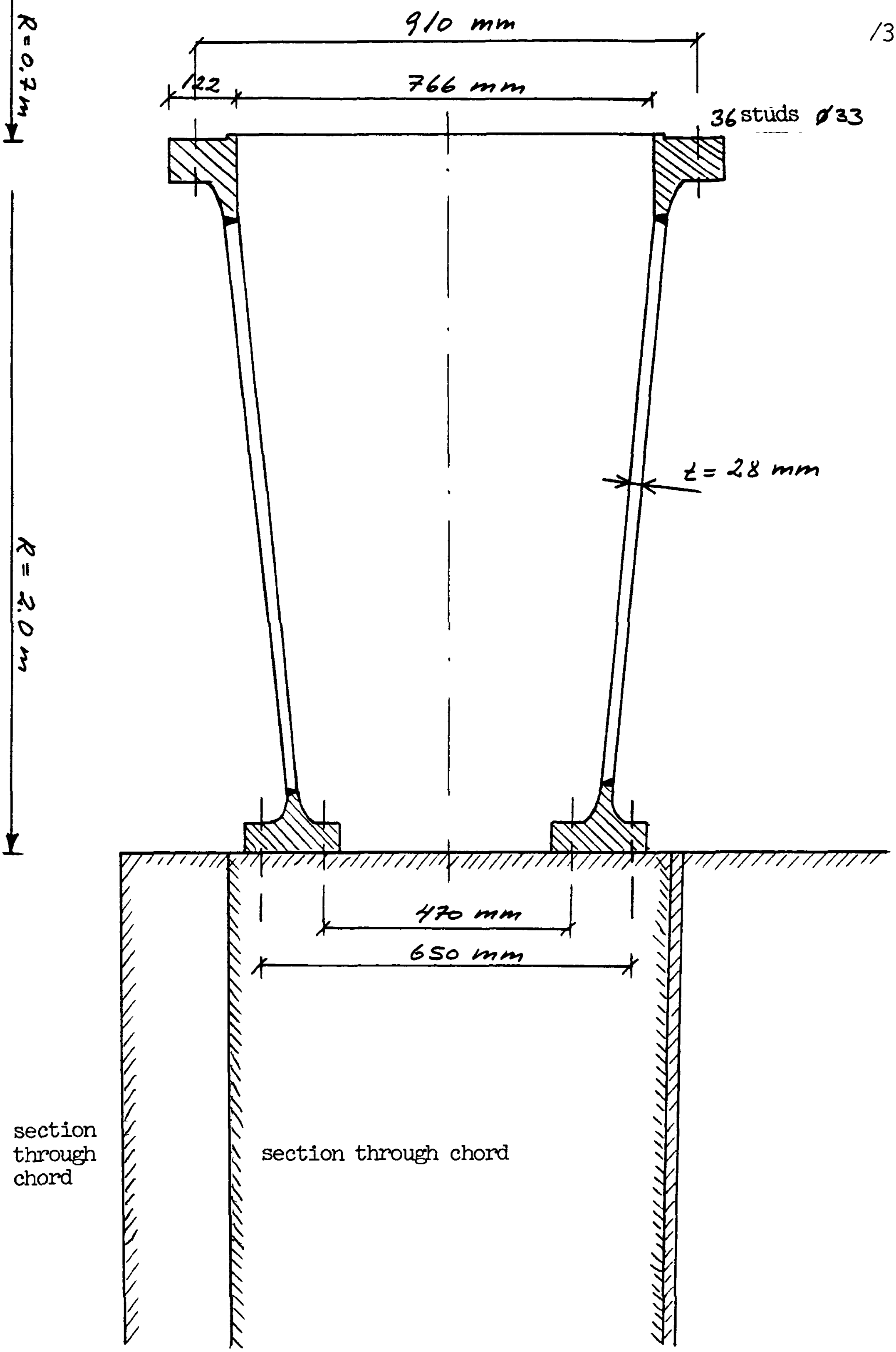


Figure 2. Alternative 2.

Scale 1:10

The calculations were made on the basis of the criteria for bending in relation to the capacity for support. However, bending in relation to type of application could perhaps also be utilized and might lead to great savings in material cost. On the basis of experiences gained from the Nibe B turbine it does not seem worth while to compromise with the safety margin.

In respect to the stress the maximum limits mentioned in the VK-79 report including its appendix regarding alternating loads have been used as typical alternating loads. In other respects the alternating loads have been the determining factor for the steel sleeve. The partial coefficient of the ultimate fatigue force is

$$f_m = 1.54$$

In respect to the load:

$$f_p = 1.3.$$

We base the estimates on an annual operation efficiency of 85%.

The alternating loads mentioned in VK-79 correspond to R = 2 m. The loads closest to the base are larger and, thus, the determining factor for the steel sleeve dimensions. At the cross section R = 0.7 m the stress is assumed to be 20% greater (but apparently within the safety limit) than at R = 2m.

3. DIMENSIONING OF THE STEEL SLEEVE /5

In the appendix of the Department of Fluid Mechanics Report VK-79 the maximum alternating loads are given for six different directions of the cross section.

Starting out from the distribution of the wind speed indicated in VK-79 and the graphs of the alternating loads in its appendix, the relative damage due to fatigue in the individual directions can be estimated according to the theory on linear partial damage.

For an optionally selected section ($W = 3 \cdot 10^{-3}\ m^3$) the partial damages will after one year amount to:

θ	$d = \Sigma \frac{n_i}{N_i}$
-30°	0.087
0°	0.078
30°	0.204
60°	0.450
90°	0.620
120°	0.304

From this it is obvious that M_{90} (in respect to the chords) is as expected the critical dimension.

The contributions to the fatigue from extreme loads are insignificant. The contribution from the starting/stopping moment is largest in a direction at right angle to the direction studied and does not contribute significantly to the determining direction.

The critical cross section is the one closest to the base, i.e. at R = 0.7 m.

As is obvious from the variations of the moments, it is quite feasible to approximate a constant range of the moment $M_{90,\,v} \doteq 193$ kNm of the cross section R = 2. At an optional t = 16 mm we obtain at the base:

$$\sigma_v = \frac{M_v}{W} \cdot 1,2 \cdot f_p$$

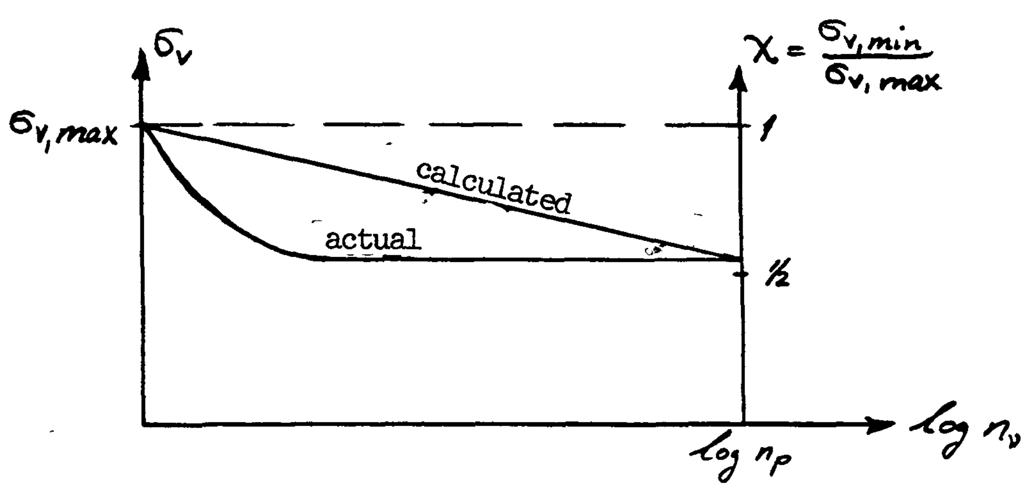
$$= \frac{193 \cdot 10^{-3} \cdot 32 \cdot 0,766}{\pi\ (0.798^4 - 0.766^4)} \cdot 1,2 \cdot 1,3$$

$$= 38,4\ \mathrm{MN/m^2}$$

According to the Danish Standards 412.2 the following is acceptable: /6

$$s_v = 73\ \mathrm{MN/m^2}/1,54 = 47\ \mathrm{MN/m^2} > \sigma_v$$

For other directions consideration must by given to the variation within the range of tension. When using the Palmgren-Miner formula (DS 412.2) dimensions too large to be acceptable appear. On the other hand the alternative method mentioned in DS 412.2 is more suitable; it is also within the safety limits since the relationships look as follows:



The requirement amounts to:

$$\sigma_{v,\,max} \simeq \Psi \cdot S_{excellent}$$

where Ψ is given in DS 412.2 and we obtain for line c

$$\underline{S_{excellent} = 47\ MN/m^2}$$

For the 6 directions we obtain:

θ	χ	ψ	M_{max} [kNm]	$W_{nec.}$ [m^3]
-30	0,5	1,0	329	$7 \cdot 10^{-3}$
0	0,28	1,1	395	$7,6 \cdot 10^{-3}$
30	0,42	1,1	405	$7,8 \cdot 10^{-3}$
60	0,78	1,0	328	$7,0 \cdot 10^{-3}$
90	0,89	1,0	301	$6,4 \cdot 10^{-3}$
120	0,90	1,0	253	$5,4 \cdot 10^{-3}$

The load factor is estimated to $3 \cdot 10^8$. /7

From this it is obvious that t = 16 is (practically) adequate since

$$\underline{W_{16} = 7,53 \cdot 10^{-3}\ m^3}$$

The maximum load occurs during the stress moment "Excessive rotation/ emergency stop" where the maximum moment amounts to ca. 660 kNm if we assume that the maximum values of both stress components occur simultaneously (this is, however, within the safety limits).

Thus, the following is acceptable:

$$\sigma = \frac{M \cdot f_{p1} \cdot f'_{p2} \cdot 1,2}{W} = \frac{660 \cdot 10^{-3} \cdot 1,2 \cdot 1,1 \cdot 1,2}{7,53 \cdot 10^{-3}} = 138\ MN/m^2$$

and

$$sf = \frac{\sigma_t}{f_m} = \frac{240}{1,54} = 156\ MN/m^2 > \sigma$$

Finally it should be stated that if it is acceptable to estimate the bending stress in relation to application as $f_m = f_p = 1.0$, the dimension could be reduced to t = 10 mm.

θ	χ	ψ	M_{max}	W_{nec} m^3
-30°	0,5	1,0	253	$3,5 \cdot 10^{-3}$
0°	0,28	1,1	304	$3,8 \cdot 10^{-3}$
30°	0,42	1,1	312	$3,9 \cdot 10^{-3}$
60°	0,78	1,0	252	$3,5 \cdot 10^{-3}$
90°	0,89	1,0	232	$3,2 \cdot 10^{-3}$
120°	0,90	1,0	195	$2,7 \cdot 10^{-3}$

$$W_{10} = 4,6 \cdot 10^{-3}\ m^3 \quad > \quad W_{necessary}$$

4. FLANGES /8

4.1 Inner Flange, R = 0.7 m

Since the inner flange must fit together with the flanges of the inner B-sleeves and since the stress is considerably less than on the present B rotor blades, a flange like the one already existing is suggested.

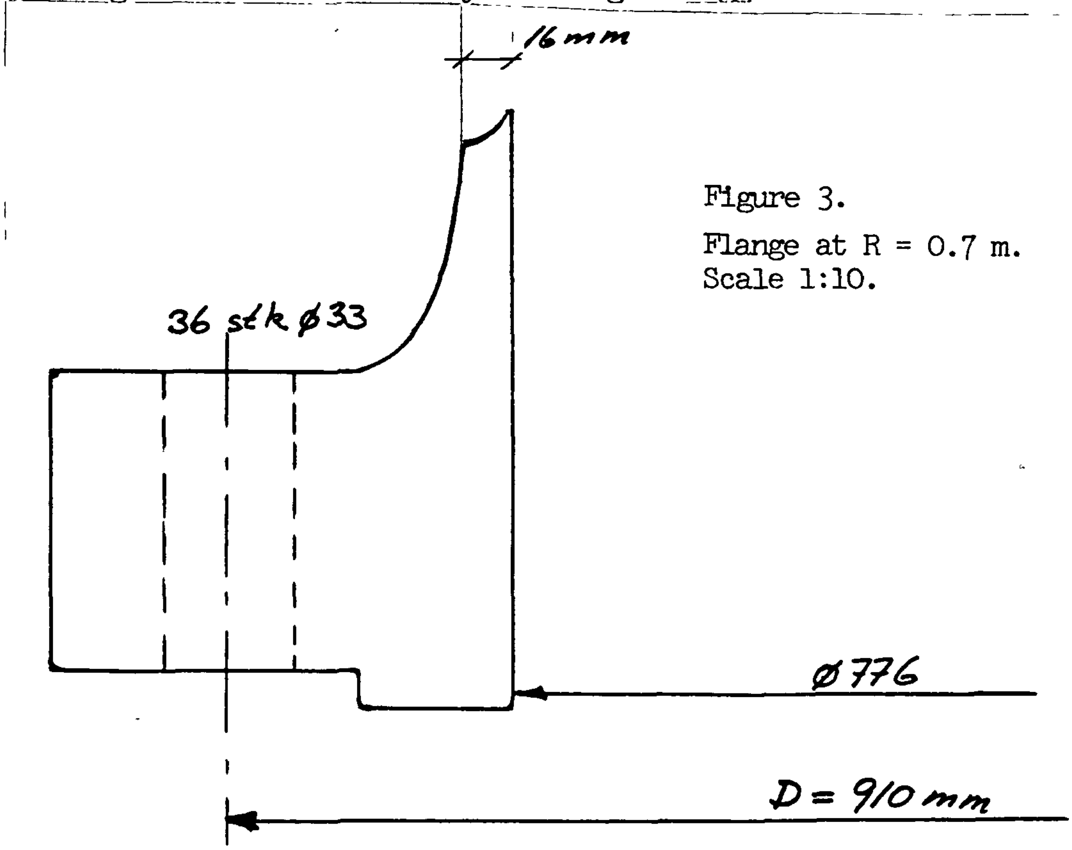


Figure 3.
Flange at R = 0.7 m.
Scale 1:10.

Although not yet the object of calculations this design is for the time being considered to possess the nesessary strength.

4.2 Outer Flange. $R = 2.0$ m

This end of the sleeve must in contrast to the one near the hub have an internal flange. The determining factor for this flange, which is a very important part, is the requirement for rigidity. Due to the eccentricity forces (between the wall of the sleeve and the studs) the sleeve is subjected to torsion. The torsion angle must be small enough to reduce the bending stress /9
on the studs glued in. In addition this part must have a certain rigidity in order to fulfill the demand that the range of tensile strength in the prestressed threaded pin must be reduced to at most 15% of the stress applied. It is, however, assumed that this requirement is automatically fulfilled if the flange is given the necessary torsion resistance.

This kind of analyses have, however, not yet been made. A finite-element analysis would be well suited for studying these conditions in more detail. Then it might also be possible to establish how large the concentration of tensile strength is in the flange, although such a tension may be without consequence.

A qualitative idea of the shape of this flange is furnished below.

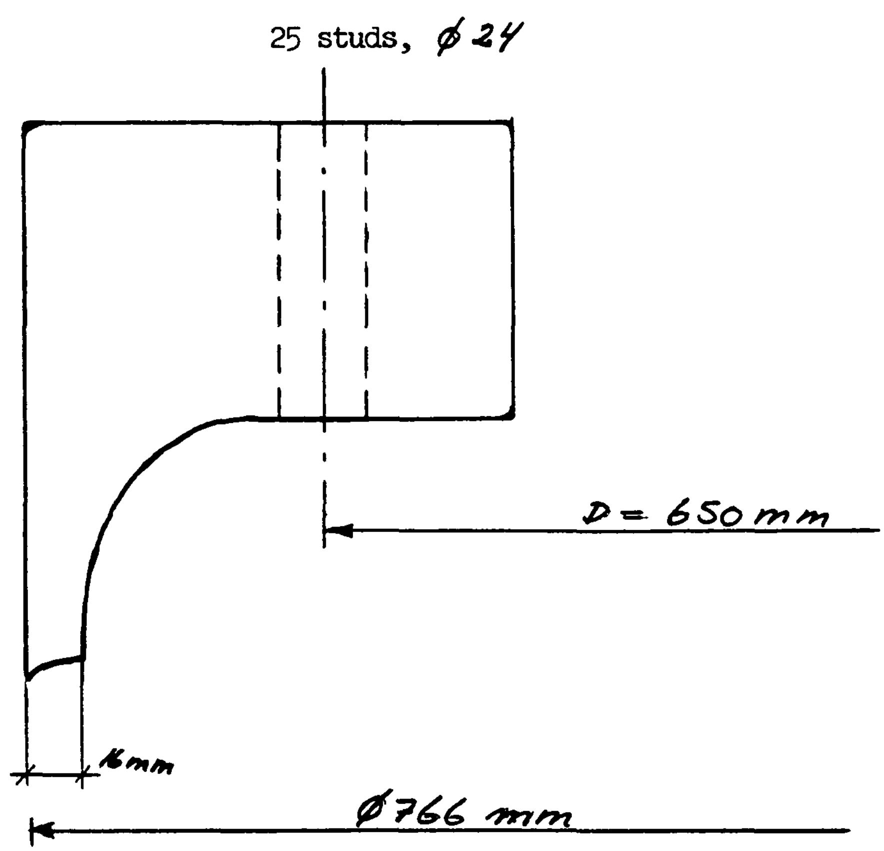


Figure 4. Flange at R = 2.0 m. Scale 1:2

REPORT IV

SHELLS OF CROSS-LAMINATED WOOD FOR ROTOR BLADES

Dan Jepsen

Logo of LNJ Spaedtrae

SHELLS OF CROSS-LAMINATED WOOD FOR ROTOR BLADES

by

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Translation of "Vingeskaller af krydslimede traelameller", LNJ Spaendtrae Co., February 1982, 2 p.

The following report is based on technical experiments concerning manufacturing procedures conducted at our own initiative at the LNJ Spaendtrae Co., 6261 Bredebro, Denmark, during December of 1981 and January of 1982. /1

The aim of the tests was to demonstrate that by gluing together ordinary wooden boards in a special "bedding" it is possible to produce a curved shell of wood with minimum expenditures of material and work. Two such shells will, when assembled, form a finished rotor blade.

The bedding is constructed of ribs at a distance of 0.5 m from each other. These ribs are cut according to the actual exterior shape of the rotor blade both as far as curvature and torsion of the chord from tip to base are concerned. See Figure 1.

Glue is applied to three sides of the laminating boards, presorted according to tensile strength and hand sorted according to desired dimensions. They are then placed close together within the bedding. The first layer is placed in parallel with the leading edge, the second layer in parallel with the trailing edge of the rotor blade, etc. The individual layers will, thus, form an angle in relation to each other and give the construction satisfactory homogenicity and strength. The thickness of the shell can be varied by making a slight

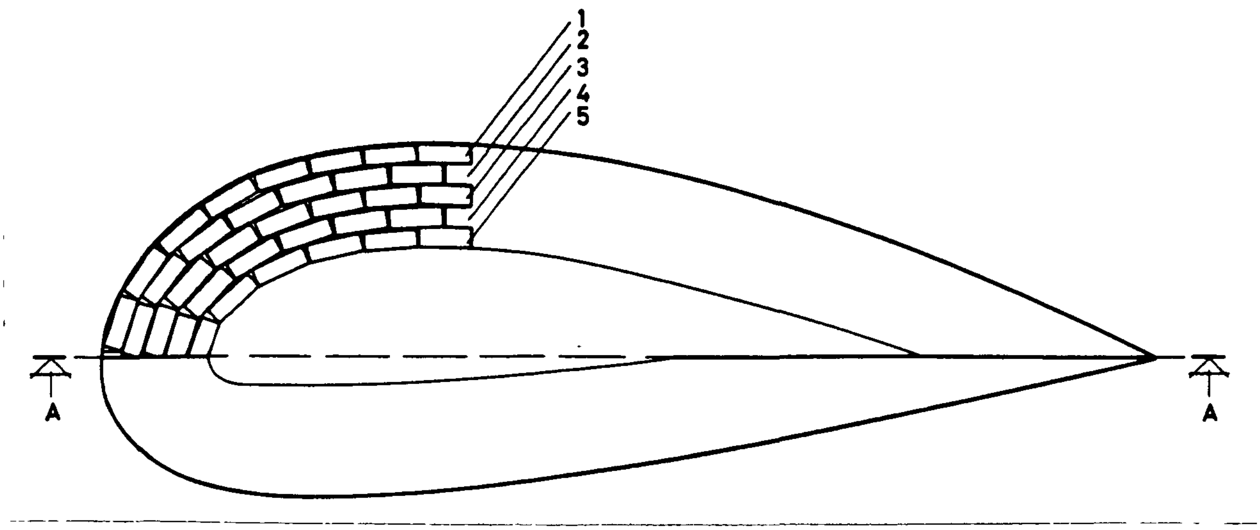

Figure 1. Cross section of rotor blade shell. /2

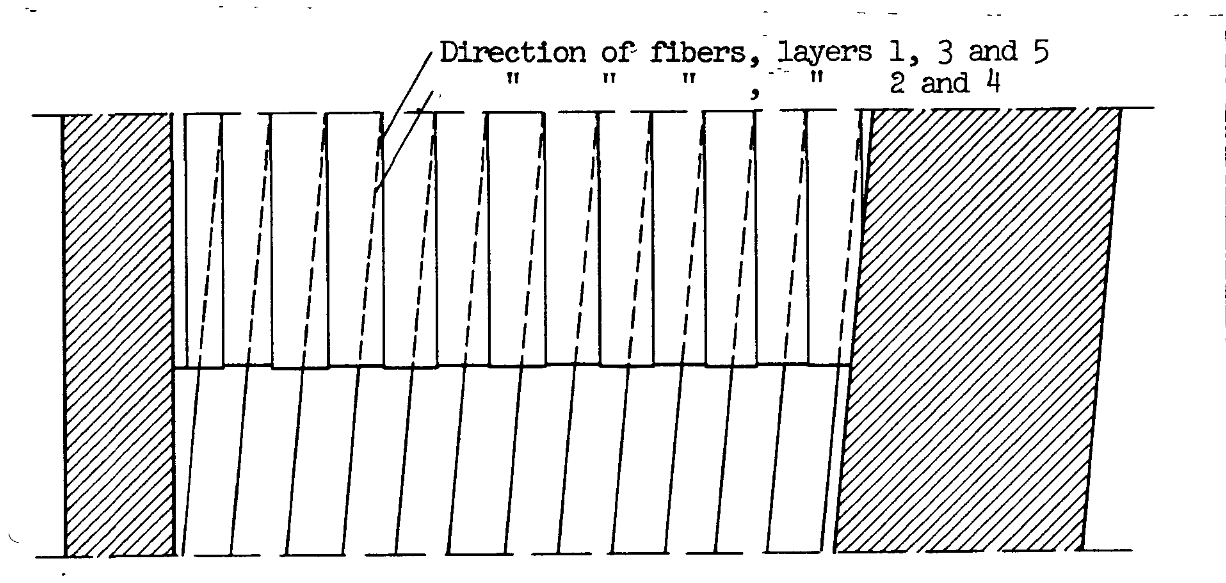

Figure 2. Detail of section A - A. /2

internal tapering longitudinally between the individual layers; see Figure 2. /1 cont.

The plan is to use a joint-filling type of resorcinol-phenol glue. The necessary contact between both the narrow and the broad sides of the laminating boards can be achieved by means of a specially constructed pressure tool applied over each rib.

When both halves of a rotor blade have been finished (each in its own bedding) the contours must be trimmed whereafter the halves can be glued together. A suitable surface for this purpose may be wooden splints placed between the two halves along the leading and the trailing edges of the rotor blade.

In connection with the joining of the halves some longitudinal ribs can be glued to the insides of the shells for taking up dislocation forces or for serving as anchors for fittings.

REPORT V

GLUED-IN STUD BOLTS FOR ATTACHMENT OF WOODEN ROTOR BLADES TO WIND TURBINES

H. RIBERHOLT

GLUED-IN STUD BOLTS FOR ATTACHMENT OF WOODEN ROTOR BLADES TO WIND TURBINES /cover p.

by

H. Riberholt

Translation of " Indlimede Bolteforbindelser til Indfaestning af vindmoellevinger", Department of Load-Bearing Constructions at the Danish Technical University, November 1981, 11 pages.

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GLUED-IN STUD BOLTS FOR ATTACHMENT OF WOODEN ROTOR BLADES TO WIND TURBINES

1. INTRODUCTION

This report contains a review of the literature, an evaluation of the test results as well as the design of the bolt connections and their suitability for attaching wooden rotor blades to wind turbines. In addition outlines are drawn up for further development and for a test program.

The report is part of the preliminary research on a project concerning wooden rotor blades for wind turbine B at Nibe, Denmark. The report has been given financial support by the Wind Power Project of the Department of Energy as well as by the Danish Electric Power Companies.

2. REVIEW OF THE LITERATURE

The following two reports concern experiments made with glued-in stud bolt connections in respect to their applicability for building constructions:

Gunnar Edlund, "I limtrae inlimmad skruv" [Stud glued into laminated wood], National Association of Swedish Wooden House Manufacturers (STFI), report B. no.333, 1975.

H. Riberholt, "Bolte indlimet i limtrae", [Studs glued into laminated wood], Department of Building Construction (ABK), report R 83, 1977.

Table 2.1 furnishes a review over the factors determining the strength having been studied according to these reports.

The following report deals among others with tests made on glued-in stud bolt connections in respect to their application for attaching rotor blades to wind turbines:

James R. Faddoul, "Test evaluation of a laminated wood wind turbine blade concept", NASA T4-81719, 1981.

In this report an attempt is made to determine the effects of the following strength-determining factors:

TEST SERIES 1.

Type of wood: solid wood - veneer
The distance to which the stud is glued in.
The design of the holes. Constant or tapering diameter.
Lateral reinforcement of wooden butt.

TABLE 2.1

	Test Parameter	Stud diam. mm	Glued-in portion mm	Nos of tests	Remarks
Studs set in parallel with direction of fibers	**Axial extension**				
	Pilot test	16 24	150 200		Type of failure in relation to dimension of test piece cross section
	Short-term test	16 24	160 320 240 480	5 5 5 5	Reference test using the manufacturing mode selected and the dimensions of the test piece.
	Yield of studs	16	150 320	5 5	Does the yield of the studs affect their resistence to tension?
	Variation in moisture content	16	160 320	2·3 2·3	Strength following 4th and 16th moisture cycles.
	Long-term test	16	160 320	6 6	
	Alternating load	16	160 320	2·3 2·3	
	Shear stress				
	Short-term test	16	160	3·3	Reference tests using different dimensions of wood
	Effect of shearing stress on axial strength	16	160	2·3	
	Axial compression				
Studs set at right angle to the direction of fibers	Short-term test	16	160 320	3 3	Reference test.
	Variation in moisture content	16	160 320	2·3 2·3	Strength following 4th and 16th moisture cycles
	Alternating load	16	160 320	5 5	

TEST SERIES 2. /2

Epoxy glue filler.
Reduction in the rigidity of the wood in relation to additional holes tapped.
Stud thread.
Studs with constant cross section/ tapering cross section.

In the report mentioned above the number of tests repeated is relatively low and in combination with the fact that a failure often occurred during the attachment of the test pieces (at the opposite end of the glued-in stud) it provides a rather insecure basis for any conclusions.

3. EVALUATION OF TEST RESULTS

On the basis of the three reports mentioned above, an evaluation of the types of glue and the designs of holes and studs is furnished below.

3.1 Types of Glue

It is preferrable to use glue which does not require pressure to set.

Glues which expand can act as reinforcement of the surrounding wood since small fissures and cracks can be filled with glue. This effect can be achieved by injecting glue into the cavity under pressure.

It has proved feasible to use resorcinol-phenol glue which needs pressure but does not adhere to steel. This requires thread along the entire distance of the stud to be glued in so that the transfer of the stress from the glue to the steel is accomplished by physical contact. The stud must in addition be inserted into a hole with a diameter slightly less than the outer diameter of the stud itself.

Edlund [1975] demonstrated clearly that polyurethane foaming glue is too ineffective for the use in overdimensioned holes. Only when using 2-component polyurethane, epoxy or resorcinol glues will the wood and not the glue fail.

Riberholt [1977] demonstrated that the static tensile strength is only moderately reduced when test pieces glued together with resorcinol-phenol glue are alternately soaked and dried out a number of times. The studs were provided with thread and screwed into a tight hole. Tests revealed that water has penetrated ca. 50 mm along the stud. The use of epoxy glue could perhaps have prevented this since epoxy in contrast to resorcinol-phenol adheres to steel. The epoxy glue itself must not be water soluble.

The diameters of the boreholes were larger by 1/8" to 3/4" than the dia- /3
meter of the studs in all the tests reported by Faddoul [1981]. This necessitated the use of an expanding glue; consequently, epoxy with fillers was used. The emphasis was placed on asbestos and charcoal fibers as fillers, of which the latter seems to be the most suitable; see Figure 2.

Figure 2 illustrates in addition that the fatigue resistance of the stud connection is strongly affected by the glue. It should therefore be an advantage to use a glue with better fatigue properties and/or design the glued seam so that the fatigue resistance becomes increased. The latter can be accomplished by reducing the width of the glue seam, i.e. by using one of the following methods:

Studs with constant diameter.

Smooth studs: Diameter of the hole = diameter of the stud + what is necessary in respect to tolerances. The stud must not strain against the sides of the hole; see, e.g., Edlund [1975].

Threaded studs: Diameter of the hole = mean outer diameter of the thread of the stud and its stem; see Riberholt [1977].

Studs with tapering diameter.

Tapering boreholes without discontinuous gradation of the bore diameter.

The effect of the width of the glue seam on the fatigue resistance can within a certain error margin be determined for differently wide glue seams by comparing the correlation between failure due to dislocation stress during static tests and failures during fatigue tests. For tapering boreholes an area of 70 square inches has been taken under consideration corresponding to the surface of the graded hole.

The data indicated in Table 3.1 seem to support the hypothesis that better fatigue properties can be achieved by using narrow glue seams than using wide ones.

TABLE 3.1 FISSURE DISLOCATING STRESS, NUMBER OF LOAD CYCLES UNTIL FAILURE /4

	1" studs, L = 15" Conic holes Faddoul [1981]		16 mm studs, $L = \frac{20+10}{2} \cdot 16$ mm Tight holes Riberholt [1977]
Static tests	Stud diam. constant	Conic studs	Thread diameter constant
Static tests	7.5 MPa	8.1 MPa**	7.1 MPa
Fatigue tests	3.4 MPa**	3.4 MPa**	4.5 MPa
No. of load cycles	$5 \cdot 10^5$ **	$1.5 \cdot 10^6$ **	$\sim 10^5$
Correlation	0.45	0.37	0.64

* Mean value of test data except for tests where the wood failed when the stud was inserted or the stud broke after a few tests only.

** Test series 2 only. Fatigue test using a maximum load of 35000 lbs.

3.2 Design of Holes for Studs

The strength of the glued-in stud connection is characterized by the compression effect at the surface of the wood as well as that at the end of the stud. Because of this the tensile strength of the stud connection does not grow in proportion to the distance to which the stud is glued in.

Compression at the surface of the wood causes shear stress which can initiate a rupture, so-called cracking. Tests have demonstrated that cracking can be avoided either by making the distance between the edges of the test piece adequately large (at least ca. 2.5 x the diameter of the stud; Riberholt, 1977) or by gluing on a reinforcing plate around the borehole in the wood butt. According to Faddoul [1981] a collar of 1/4" birch plywood was used. The problem is, however, hardly solved by gluing one sheet of plywood over a large butt of laminated wood. The sealing of the wood at right angle to the direction of the fibers could lead to cracks in the butt end of the laminated wood. It might be better to glue either several smaller plywood plates just around the studs or a collar corresponding to the circle of studs.

Faddoul [1981] demonstrated that the static tensile strength can be increased if for the same diameter of stud a conic hole is used instead of a straight hole with constant diameter. The mean value obtained during two tests on 15" studs and straight holes was 63,600 lbs but for conic holes 76,400 lbs., i.e., a 20% increase in strength. This should be viewed from /5
the point of view that the area between glue and wood is increased from 47 square inches to 70 square inches, i.e. by 49%.

The effect of a tapered hole on the fatigue limit cannot be determined on the basis of the tests made. It should, however, be emphasized that stepwise changes in the hole dimension most likely reduces the fatigue resistance of the glue seam. It should therefore be investigated in what manner a tapered hole should be tapped and what the fatigue resistance then will amount to.

By means of proper design of the boreholes the compression effects can be reduced. When making both stud and hole tapered, the compression effect inside the wood will be reduced. However, the compression effect at the surface of the wood can hardly be very much reduced. Therefore it does not mean so much that the compression inside the wood is reduced. The idea of using additional holes for taking up the stress cannot be evaluated on the basis of the two tests reported by Faddoul [1981].

3.3 Design of Studs

The contour of the stud to be glued in can be either smooth or threaded as long as in the case of a smooth contour a glue is used which is able to adhere to steel.

Studs have been used for the building industry which have a coarse pitch thread and are screwed into holes with such a small diameter that there is physical contact between steel and wood. This physical contact gives a certain

strength to the connection in the case that the glue seam should fail entirely.

During the manufacture of wind turbine rotor blades it should be expected that the production control would always assure perfect gluing so that smooth studs can be glued securely into oversized holes. It is of course possible to replace smooth studs with threaded. Then the requirements on cleaning the hole can be diminished.

According to Edlund [1975] and Faddoul [1981] about the same strength of the glue seams when setting studs in overdimensioned holes can be accomplished independently of the design of the stud contour. Faddoul [1981] has, however, demonstrated that for studs with tetragonal thread (5 UNC rounded acme THD) fatigue failures of the epoxy glue occur frequently, yet without being of any major consequence.

According to Table 3.1 it seems that greater strength is achieved when using conic studs in graded holes than when using studs with a constant diameter. This pertains to both static and fatigue limit tests. However, due /6
to the few tests made, the difference is not statistically significant.

The axial strength of the stud depends on the design of its contour. No failures due to fatigue occurred when using conic studs during Test series 2 (see Figure 2). They are of course without thread at the neck part in the surface of the wood. There exists in addition much experimentally won information on the fatigue limit in respect to threads.

It should be contemplated whether the axial yield strength of the studs must be less than the tensile strength of the glue seam. Up to a certain point it seems assured that a difference when tightening the studs will not cause a so-called "zipper effect" affecting the entire group of studs. However, much care should be taken to assure uniform tightening.

4. SUGGESTION FOR FURTHER DEVELOPMENT OF THE STUD BOLT CONNECTION AND THE TESTING OF IT

Two geometrically essentially different designs of stud connections exist:

1. Constant diameters of both holes and studs.

2. Both holes and studs are tapered.

Both designs require glue seams which are thin, i.e. on the order of 1/10 mm. The tapered solution can be expected to have greater static strength and a higher fatigue limit than the corresponding seam with a constant diameter. On the other hand it can also be expected that these stud connections can be made equally strong just by increasing the depth to be glued by 10 - 30% in the case of constant diameter.

It is desirable that the contours of the studs to be glued in are not smooth. For a design with constant diameter it seems easy to let the thread of the projecting end of the stud continue into the glue seam. In the case of the

tapering design there is more freedom to decide whether the glued in portion should have thread or something similar.

Figure 3 illustrates two types of suggestions. Both are based on the facts that the holes are overdimensioned in relation to the studs and that the portion to be glued in is about 20 to 30 times the diameter of the stud. /7
Epoxy with a filler is suggested as glue material.

The fact that the holes are oversize should allow a reduction of the tolerances relating to the placement of the studs. The difference between the diameters of the holes and those of the studs must be determined so that there are satisfactory conditions for the gluing operation everywhere, also in the case where stud or hole is tilted, which could cause that the stud strains sharply against the side of the hole.

For each type of stud connection we suggest that experiments on static tensile strength shall be made while using two different portions of the stud to be glued in, each test to be repeated five times. The glued-in portions shall correspond to the maximum and minimum data in Figure 3. On the basis of this, the portion to be glued in can be selected and the fatigue properties of the connections tested.

The tests on fatigue limit are planned in the form of tensile strength tests using three different levels of stress. Minimum force shall be ≈ 0 and maximum force $\approx$ 40, 50 or 60% of the static short-term force, each test to be repeated five times. The stress levels suggested should be comparable to those which will occur at the attachment of the rotor blade. It is possible to supplement these tests by using other maximum and minimum values.

In total at least 2 x 5 static tests and 3 x 5 fatigue tests shall be performed for each type.

5. ANALYSIS OF STRENGTH OF A CIRCULAR GROUP OF STUD BOLTS

According to Faddoul [1981] it is assumed that in a circular group consisting of N glued-in studs the maximum resistance of a stud to a moment M can be established by means of the formula:

$$P_{stud} = 2 \cdot M/N \cdot R$$

where R is the radius of the circle of studs. It is not possible to verify this simple formula on the basis of the test results referred to by Faddoul /8
[1981]. A strain gage is, however, mentioned in connection with the glued-in studs, indicating that perhaps there are unpublished measurements which could elucidate the reliability of the above formula.

Since the formula assumes that "plane cross sections will remain plane" there must be a definite basis for its validity, i.e. that the rotor blade must be bolted to a very rigid flange. Should this flange not be "definitely rigid" it will become deformed where the studs are most strained so that these become releaved of stress. It seems therefore possible that the formula may be within the safety limit; this remains, however, to be verified.

6. ILLUSTRATIONS

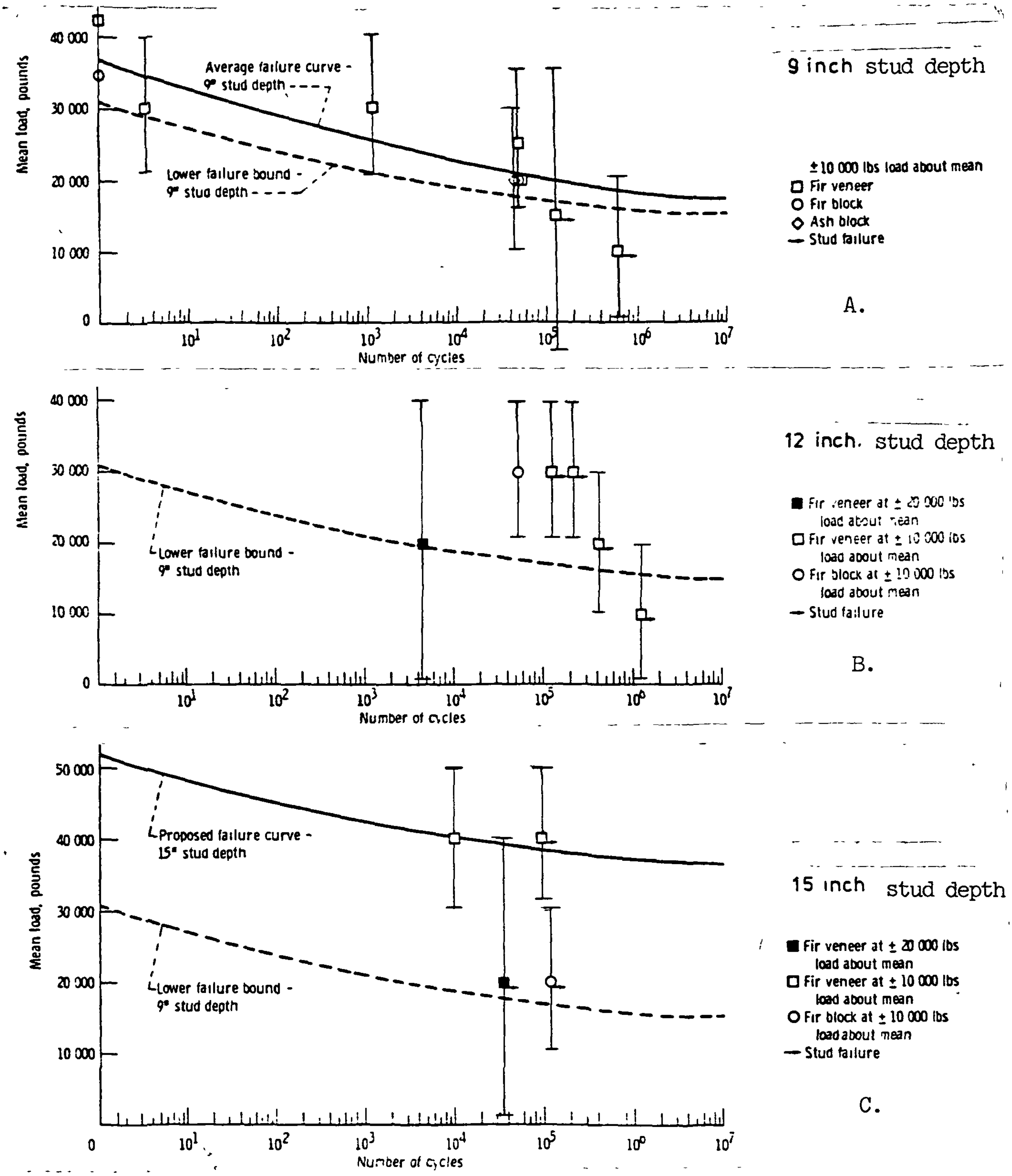


Figure 1. Fatigue limit tests with 1"studs with constant diameter (Figs. 8, 9 and 10 according to Faddoul, 1981).
A. Fir veneer, static tests, mean = 42,000 lbs.
B. Fir veneer, static tests, mean = 58,000 lbs.
C. Fir veneer, static tests, mean = 66,000 lbs.

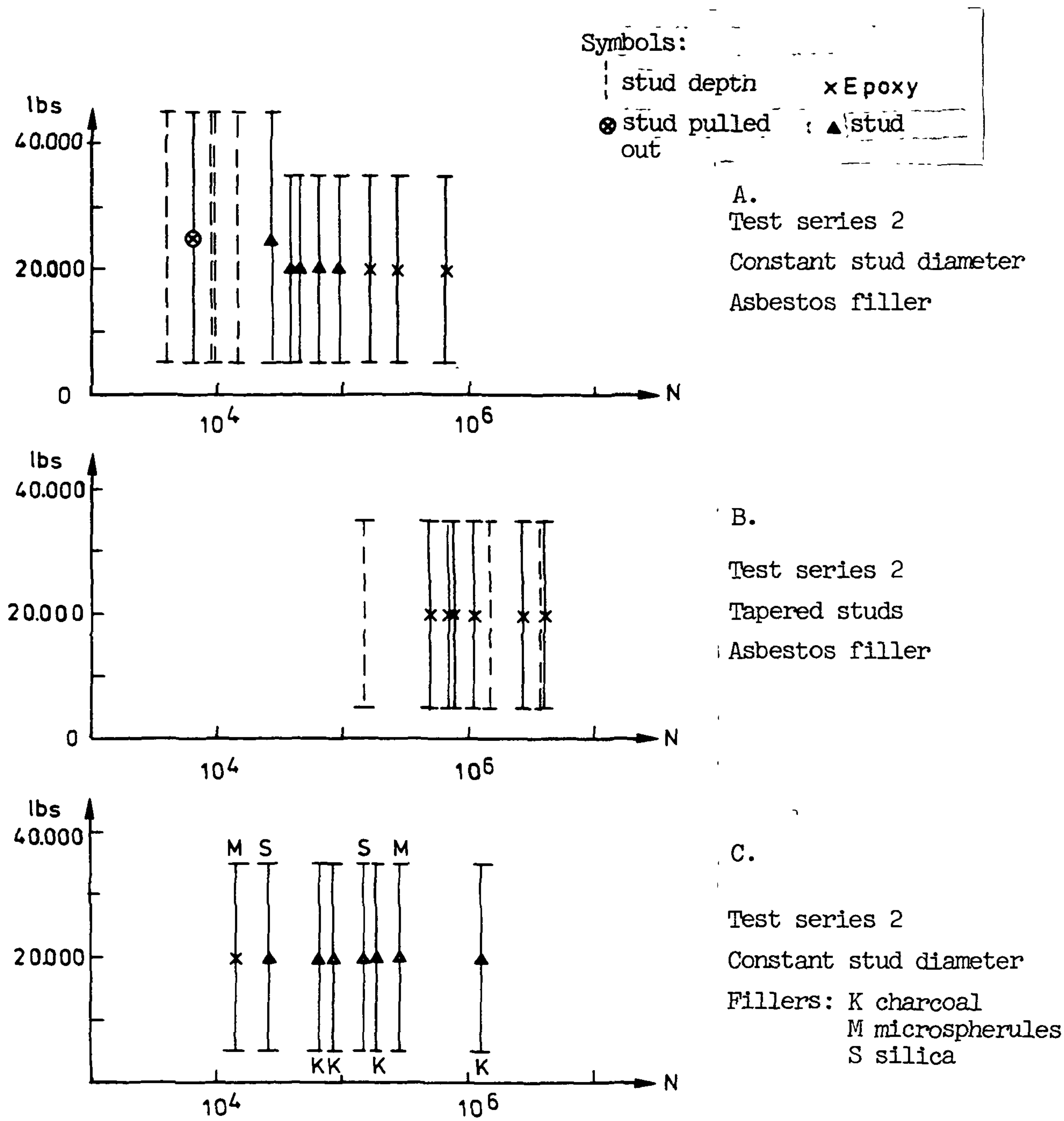
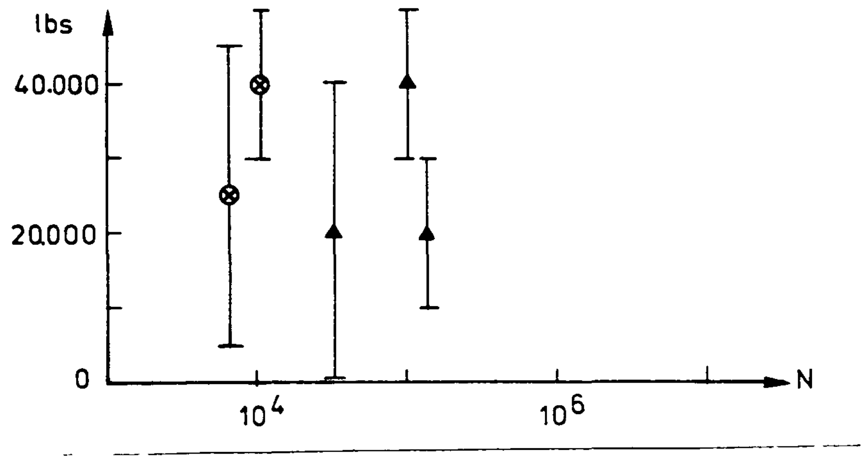


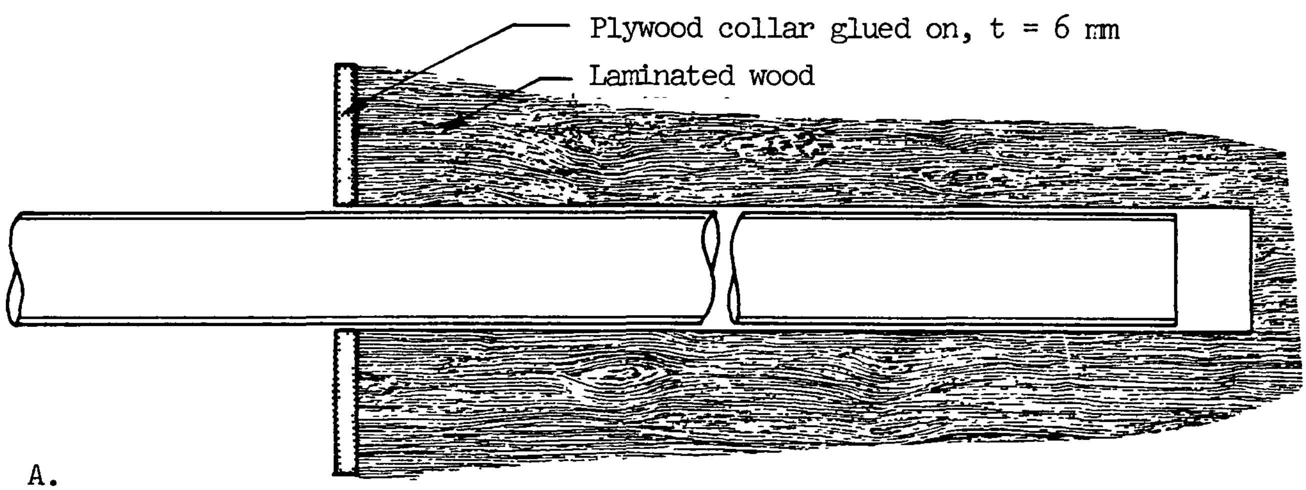
Figure 2, A – C. N number of load cycles until failure. Test series 2 according to Faddoul [1981]. (Fig. 2, D: see p. 65).



D.

Test series 1
Constant stud diameter
Constant hole diameter

Figure 2, D. N number of load cycles until failure. Test series 1 according to Faddoul [1981].



A.

Figure 3, A. Suggestions regarding stud connections.
Stud diameter D: 16, 20 or 24 mm
Diameter of hole: D + ca. 0.5 - 1 mm (depending on tolerance)
Distance glued in: ~ 20 - 30 · D
Glue: Epoxy plus filler
Thread: ISD metric according to Danish Standards 976.2, rolled on
Stud quality: 5.6, 5.8 or up to 8.8.

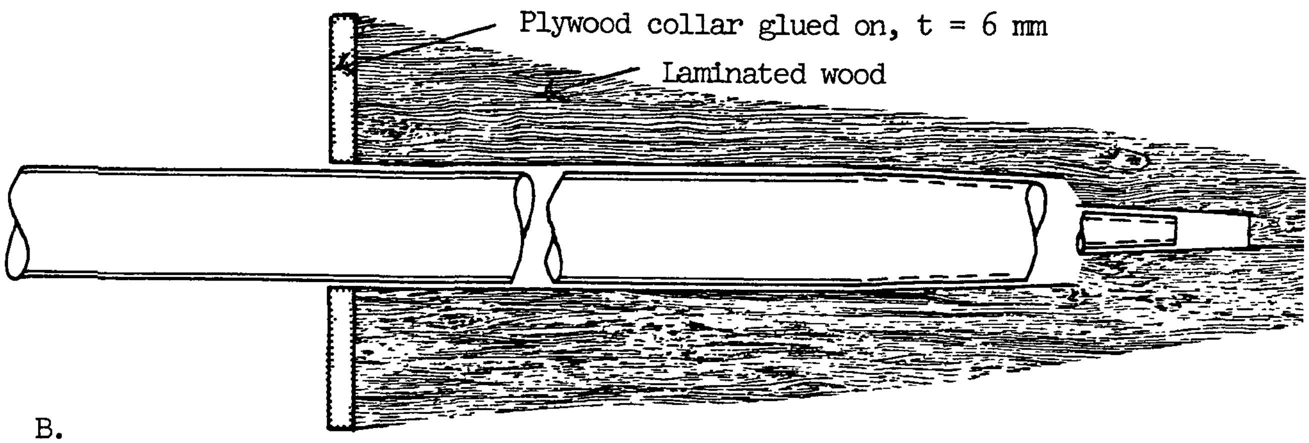


Figure 3, B. Suggestions regarding stud connections.

Stud diameter D: 16, 20 or 24 mm

Diameter of hole: D + ca. 0.5 - 1 mm (depending on tolerance)

Distance glued in: ~ 15 - 20 · D

Glue: Epoxy plus filler

Thread: ISO metric according to Danish Standards 976.2, rolled on

Tapered surface of stud: Thread or similar

Quality of stud: 5.6, 5.8 or up to 8.8.

REPORT VI. /cover p.

SUGGESTIONS FOR THE ATTACHMENT OF WOODEN ROTOR BLADES TO WIND TURBINES

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SUGGESTIONS FOR THE ATTACHMENT OF WOODEN ROTOR BLADES TO WIND TURBINES

by

H. Riberholt

Translation of "Forslag til indfaestning af vindmoelle-vinger af trae", Dept. of Load-bearing Constructions at the Danish Technical University, February 1982, 7 pages.

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1. WOODEN ROTOR BLADES, GLUED-IN STUDS /1

It is suggested that the attachment be accomplished by means of studs glued into and arranged in a circle on the butt end.

Diameter of circle:	D_k	= 650 mm
Number of studs glued in	N	= 25
Distance between studs, c - c		= 82 mm

The stud to be glued in should be designed as illustrated below:

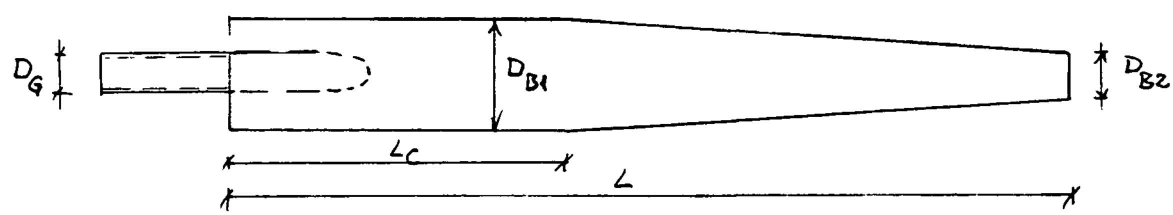


Figure 1. Dimensions of the stud. D_B = diameter of stud, L = length, D_G = threaded pin.

The stud to be glued in should be designed somewhat like a carrot, i.e. cylindrical along a distance L_c and tapering over the distance L minus L_c. At the top of the stud, a threaded hole is tapped so that a threaded pin, if necessary of a different kind of steel, can be inserted.

The steel flange at the hub is bolted on by means of the threaded pins which must be prestresses since the flange presses against the top of the stud. Any differences in dimensions between stud and flange shall be filled by plates.

Stud dimensions:

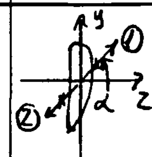
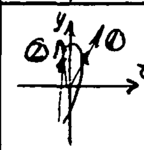
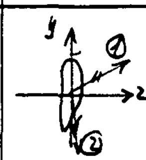
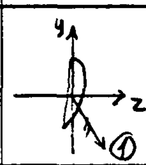
L = ?	D_{B1} = 40 mm
L_c = 1/3 L	D_{B2} = 10 mm

Diameter of threaded pin: D_g = 24 mm

2. FATIGUE RESISTANCE

According to the Department of Fluid Mechanics (AFM) Report VK-74-811201 the circle of studs is typically affected by the maximum moments mentioned below. Normal thrust and shear forces can be neglected since they are insignificant. /2

TABLE 1.

Operation Condition	Moments in [kNm] M_y	M_z	α		$\lvert M_{res}\rvert$	P_{stud} [kN]
D 1.3 light wind 6 m/sec.	20 -30	94 -85	12 199		96 ~ 100 ① 90 ~ 100 ②	25 -25
D 3.3 Wind 15 m/sec.	260 129 -	126 small	64 ~100		289 ~ 290 ① ~ 140 ②	71 34
D 5.3 Strong wind 22 m/sec.	59 -174 +	133 small	24 280		145 ~ 150 ① ~ 190 ②	37 47
S 2.3 Emergency stop during normal operation	-328 ~ 0	172 ~ 0	-62		370 ~ 370 ① ~ 0 ②	91 0
Hurricane					524	129

The effects of these moments on the point of attachment are described in the Department of Fluid Mechanics report VK-79-820105 and its appendix. In the appendix different maximum and minimum moments in various directions are indicated in relation to different operation conditions which allows us to evaluate the fatigue limit of selected glued-in studs. The studs selected are placed at right angles to the momental vectors.

It is obvious that the range of the moment = $M_{max} - M_{min}$ is considerably less than 200 kNm; only in the case of wind speeds above 20 m/sec will it be larger. The above mentioned operation condition D 1.3 furnishes, thus, an upper limit for the stud stresses during wind speeds below 20 m/sec. The VK-79 appendix states that the range of this moment at 20 m/sec may reach 250 kNm.

The stud strength, P_{stud}, is determined as follows:

$$P_{stud} = \frac{2 \cdot |M_{res}|}{N \cdot R} = \frac{4 \cdot |M_{res}|}{N \cdot D_{circle}}$$

$$= 0{,}246 \cdot |M_{res}|$$

The studs placed at $\alpha \sim 20 + 90 = 110$ are affected by many load cycles. Case I:

$$D\ 1.3 \Rightarrow P_{stud} = \pm 25 \text{ kN}$$

$$D\ 3.3 \Rightarrow P_{stud} = \begin{matrix} \sim 71 \cdot \cos(64-20) = 51 \text{ kN} \\ \sim 0 \qquad\qquad\qquad \sim 0 \text{ kN} \end{matrix}$$

This means that in both cases the strength of the studs ranges ~ 50 kN.

Studs placed at $\alpha \sim -70 + 90 = 20^0$ are affected by large forces. Case II:

$$D\ 5.3 \Rightarrow P_{stud} = \begin{matrix} 47 \text{ kN} \\ 0 \text{ kN} \end{matrix}$$

$$S\ 2.3 \Rightarrow P_{stud} = \begin{matrix} 91 \text{ kN} \\ 0 \text{ kN} \end{matrix}$$

Case II can be ignored from the point of view of fatigue. "S 2.3" and "hurricane" are not taken into consideration in respect to the stress collective; cf. Standards of Building Constructions (StBK) N2 3:34K. Thus, only case I remains having a range of stud strength amounting to ca. 50 kNm.

2.1 The Threaded Pin

In respect to the prestress, i.e. 70% of the stress necessary for failure, the range of resistance of the threaded pin will be reduced to about 15% of the force applied. The range of the stress on the threaded pin will amount to:

$$\sigma_v = 0,15 \cdot N/A_s$$

$$= 0,15 \cdot 50 \cdot 10^3/360$$

$$= 20,8 \text{ MPa}$$

In the case where the course of the stress is time constant, the factor σ_v = constant indicates according to Standards of Building Constructions N3, 1976 that the range of tension in respectto $N \geq 10^{+7}$ is

$$\sigma_{v,\ permissible} = 19.2 \text{ MPa.}$$

This means that if the range of resistence of the threaded pin is reduced by only a little, the connection will be in agreement with the requirements according to Swedish Standards and it will therefore be well within the safety limit. Another possibility is that rolled on thread would have a greater fatigue resistance. /4

2.2 The Stud Tip

This cross section is affected by the full range of force.

$$A = \frac{\pi}{4}(40^2 - 24^2) = 804 \text{ mm}^2$$

$$\sigma_v = 50 \cdot 10^3/804 = 62 \text{ MPa}$$

The Standards of Building Constructions N2 [1974] states that the permissible range of stress on steel with a rolled surface ($K_x = 1.5$) and N- 10^{+7} amounts to

$$\sigma_{v,\ \text{permissible}} = 64 \text{ MPa}$$

It should be estimated whether the condition $K_x = 1.5$ is fulfilled when the hole is tapered and rounded off at illustrated in Figure 1. Problems are not avoided when only the diameter of the stud is increased since according to the present suggestions, the distance between the stud surfaces is only 82 - 40 = 42 mm. If the studs are placed closer together it can be expected that the strength of the glue seam will be reduced. It may therefore be necessary to make the glued-in portion and the threaded portion in one piece.

2.3 The Glue Seam

The hole should be made with the same dimensions as the stud and a glue seam as thin as possible should be strived for. A 1 mm thick glue seam appears to be satisfactory. As glue, epoxy plus filler is suggested.

The glued-in surface illustrated in Figure 1 amounts to

$$A = \tfrac{1}{2}\pi D_{B1} L\left[1 + \frac{L_c}{L} + \frac{D_{B2}}{D_{B1}}\left(1 - \frac{L_c}{L}\right)\right]$$

L =	300	400	500	600	mm
A =	28,3	37,7	47,1	56,5	10^3 mm^2

$$\tau_{max} = P_{stud}/A$$

See also Table 2 on page 72.

Table 3 illustrates the results of tests indicating the correlation between /5
τ and the expected number of load cycles (N) leading to a failure.

It is suggested that a test series on the fatigue limit shall be made while using a glued-in portion of the stud of 500 mm. This should provide a satisfactory safety margin in respect to failure as long as the stud is tested for a load spectrum corresponding to Case I or Case II or a combination thereof and while N is on the order of 10^7.

TABLE 2. MAXIMUM DISLOCATING STRESS MPa

Op. condition	L = 400 mm	600 mm	
D 1.3	0,66 -0,66	0,44 -0,44	Case
D 3.3	1,35 0	0,90 0	I
D 5.3	1,25 0	0,83 0	Case
S 2.3	2,41 0	1,61 0	II
Hurricane	3,42	2,28	

TABLE 3. CORRELATION BETWEEN τ AND THE EXPECTED NUMBER (N) OF LOAD CYCLES

Source	Stud			τ [MPa]	N
Faddoul	Constant diameter	$D_{B1}=1"\sim 25$ mm	L=15"	3,4	$5 \cdot 10^5$
	Tapering "	$D_{B1}=1\frac{3}{8}"\sim 35"$	L=15"	3,4	$1,5 \cdot 10^6$
Riberholt	Constant diameter	$D_{B1}=16$	L=10D	4,6	10^5
		$D_{B1}=16$	L=20D	3,1	10^5

3. RESISTANCE UNDER CONDITIONS OF STARTING/STOPPING OR EXTREME LOADS /6

In spite of the fact that there is of course a combination of operating and extreme stresses which can result in failure, they will here at first be treated separately.

According to the Department of Fluid Mechanics report VK-79 the momental range due to Starting/Stopping during light wind conditions amounts to maximum 120 + 90 = 210 kNm, i.e. corresponding to condition D 1.3.

Starting/Stopping during strong wind conditions as well as Emergency Stop when running at normal speed amount to extreme values of M_1 and of the stud resistance P equal to:

$M_{min} = -310$ kNm $M_{max} = 200$ kNm

$P_{min} = -76$ kN $P_{max} = 49$ kN

The corresponding stress (MPa) on the thread, the reduced area of the stud and the glue seam will amount to

σ_{thread}	-32	20	
$\sigma_{red., stud}$	-94	61	
τ_{glue}	-1,61	1,04	(L = 500 mm)

Since the frequency when this may occur amounts to 500/year it appears that the strength is adequate because the collective stress on the thread results in a p-value of ca. 1/3 and, thus, within a satisfactory range of stress, i.e. 55.2 MPa, in relation to N 10^7. Correspondingly, for the reduced cross section of the stud a permissible range of stress in case of $K_x = 1.5$ is 178 MPa.

Similarly, the following data can be obtained for extreme loads:

TABLE 4.

	25% excess. rotation + emerg. stop		hurricane	
M kNm	-485	450	-365	525
P kN	-119	111	- 90	129
σ_{thread} MPa	- 50	46	- 38	54
$\sigma_{red.,stud}$ MPa	-148	138	-112	160
τ_{glue} MPa	-2,5	2,4	-1,9	2,7

On the basis of the previously mentioned fatigue forces in respect to the glue seam, it appears that this will be adequately strong.

As far as the steel component is concerned, the Swedish Standards state that extreme load cycles amounting to less than 100 occasions can be negelcted.

CALCULATION OF ALTERNATING LOAD ON THE WOODEN ROTOR BLADE
OF WIND TURBINE B AT NIBE

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CALCULATION OF ALTERNATING LOAD ON THE WOODEN ROTOR BLADE
OF WIND TURBINE B AT NIBE

by

Stig Oeye

Translation of "Veksellastberegning for Nibe B traevinge", Department of Fluid Mechanics Report VK-79-820205, edition 1, February 5, 1982, 14 pages. Distribution: Project T.

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SUMMARY

On the basis of revised mean data dynamic calculations have been made concerning the wooden rotor blade of wind turbine B at Nibe, Denmark, during a large number of operating conditions. Graphs have been drawn of maximum and minimum data regarding bending stresses at a number of cross sections of the rotor blade as a function of wind speed and the amount of expected numbers of load cycles.

1. MEAN DATA, ETC. /1

The mean data utilized are illustrated in Figure 1. In relation to the previously used data on the wooden rotor blade some adjustments have been made in that we now take a trailing edge made of 12 mm plywood and an elasticity module of the laminated wood amounting to 13.5 GPa into consideration. The weight of the rotor blade including the steel sleeve and the flanges at the attachment point amounts to about 2700 kilos. The frequencies of natural vibration at operating speed can be estimated to:

Vibration	ω rad/s	ω/Ω "p"
1. flap	15,4	4,4
1. chord	37,2	10,6
2. flap	43,5	12,4

2. CALCULATION OF ALTERNATING LOAD DURING NORMAL OPERATION

The forced response of the rotor blade during a number of operating conditions were calculated by means of the rotor simulating program "FIX": for each of the wind speeds 6, 10, 13, 18 and 25 m/sec, cases were calculated where the yaw angles are -15^{o}, 0^{o} and $+15^{o}$ (i.e., with the turbine positioned at 15^{o} to the left of the wind, directly into the wind and at $+15^{o}$ to the right of the wind direction). All the cases were calculated for a wind gradient (exponent = 0.18) and a "tower shadow". For wind speeds up to 13 m/sec the pitch angle is 0^{o} but for greater wind speeds the turbine must be positioned so that the axial effect of the three rotor blades equals 700 kW. The results of these calculations are presented in the form of time sequences over 3 rotor revolutions in relation to the bending moments at a number of cross sections of the blade. Within each of the sections the bending stress on each flap (M_1) in the direction of the local chord (considered to be positive when bending backward in the direction of the wind) and the bending stress on each chord (M_2) at right angle to the chord (considered to be positive when bending forward in the direction of the rotation) are calculated. At the cross section R = 2 m (the attachment) the direction of the axes are, however, parallel and at right angle, respectively, to that of the tip chord. Thus, e.g., in Figure 2 the moments at R = 2 in relation to V = 13 m/s and yaw = $+15^{o}$ are illustrated. It can be seen that a distinct minimum occurs during each revolution.

Figures 3 - 10 are drawn up on the basis of these calculations so that both graphs in each figure illustrates the absolute maximum and minimum observed within yaw = $\pm 15^{o}$ as a function of the wind speed. During normal operation these moments will stay between the graphs. The calculation of the fatigue limit, based on the full variation of moments between the extreme data presented for a given wind speed at one single load cycle per revolution, should therefore be expected to furnish results within the safety limits. /2

3. DISTRIBUTION OF WIND SPEED

Since at Nibe the wind speed at 45 m elevation is assumed to have a Rayleigh distribution with a mean wind speed of 8 m/sec, the following distributions of hours of operation and, thus, also of rotor blade revolutions (load cycles) per year are obtained and distinguished into 2 m/sec speed intervals:

Wind , m/s	hours/yr	revol./yr
5 - 7	1647	$3{,}3 \cdot 10^{6}$
7 - 9	1559	$3{,}1 \cdot 10^{6}$
9 - 11	1253	$2{,}5 \cdot 10^{6}$
11 - 13	885	$1{,}8 \cdot 10^{6}$
13 - 15	552	$1{,}1 \cdot 10^{6}$
15 - 17	298	$6{,}0 \cdot 10^{5}$
17 - 19	149	$3{,}0 \cdot 10^{5}$
19 - 21	70	$1{,}4 \cdot 10^{5}$
21 - 23	26	$1{,}0 \cdot 10^{5}$
23 - 25	9	$1{,}8 \cdot 10^{4}$
Total	6447	$1{,}3 \cdot 10^{7}$

4. SEQUENCES OF STARTING/STOPPING /3

In connection with the starting and the stopping of the wind turbine the rotor blades are subjected to great stresses somewhat deviating from those during normal operating conditions. These moments can be further subdivided into startin/stopping during light wind and startin/stopping during strong wind as well as emergency stopping at normal speed.

The latter two moments are somewhat similar.

The following loads and frequencies can be expected:

STARTING/STOPPING DURING LIGHT WIND
Number of load cycles: $2 \cdot 10^4$/year.

R m	M_1 kNm min	M_1 kNm max	M_2 kNm min	M_2 kNm max
2	- 120	90	- 90	90
6	- 70	38	- 38	38
10	- 26	12	- 12	12
14	- 6	3,5	- 3,5	3,5

STARTING/STOPPING DURING STRONG WIND + EMERGENCY STOP
DURING NORMAL SPEED
Number of load cycles: 500/year.

R m	M_1 kNm min	M_1 kNm max	M_2 kNm min	M_2 kNm max
2	- 310	200	- 90	150
6	- 195	145	- 38	75
10	- 95	78	- 12	31
14	- 31	27	- 3,5	9

5. CASES OF EXTREME LOADS

For the sake of completeness three cases of extreme load moments will be included which the rotor blade must be able to withstand. The first two, i.e. 25% excessive rotation at pitch = 0^o and strong wind followed by an emergency stop, belong evidently to a single extreme load cycle where the duration of the maximum load is on the order of 1 second. The case during a hurricane corresponds to an extreme gust of wind and can be expected to have a duration of several seconds. The load moments should therefore be correlated with short-term strength of the rotor blade. /4

25% EXCESSIVE ROTATION + EMERGENCY STOP
Number of load cycles: 1/yr

R m	M_1 kNm min	M_1 kNm max	M_2 kNm min	M_2 kNm max
2	- 485	450	- 90	235
6	- 305	320	- 38	115
10	- 145	165	- 12	48
14	- 48	57	- 3,5	14

DURING HURRICANE WHEN NOT OPERATING
Number of load cycles: 0.1/yr

R m	M_1 kNm min	M_1 kNm max	M_2 kNm min	M_2 kNm max
2	- 365	525	relativt små	
6	- 200	287		
10	- 90	130		
14	- 29	41		

6. TABLE AND GRAPHS

N	X (M)	EI1 (NM2)	EI2 (NM2)	GJ (NM2)	M (KG/M)	KM2 (M2)	BETA (R)	BEM (R/M)	T (N)
	0.70								
1		6.00'+08	6.00'+08	4.60'+08	5.70'+02	1.60'-01	0.00'+00	0.00'+00	1.68'+05
	2.00								
2		3.74'+08	7.00'+08	6.00'+07	4.50'+02	2.27'-01	-1.56'-01	1.00'-02	1.56'+05
	3.00								
3		2.59'+08	8.73'+08	4.39'+07	2.62'+02	2.10'-01	-1.48'-01	1.00'-02	1.42'+05
	4.00								
4		1.73'+08	6.96'+08	3.08'+07	2.19'+02	2.04'-01	-1.40'-01	1.00'-02	1.31'+05
	5.00								
5		1.10'+08	5.52'+08	2.07'+07	1.81'+02	1.97'-01	-1.33'-01	1.00'-02	1.19'+05
	6.00								
6		6.66'+07	4.37'+08	1.33'+07	1.46'+02	1.92'-01	-1.25'-01	1.00'-02	1.07'+05
	7.00								
7		3.75'+07	3.41'+08	8.00'+06	1.14'+02	1.90'-01	-1.17'-01	1.00'-02	9.55'+04
	8.00								
8		2.31'+07	2.72'+08	5.25'+06	9.46'+01	1.82'-01	-1.08'-01	1.00'-02	8.51'+04
	9.00								
9		1.78'+07	2.23'+08	4.06'+06	8.41'+01	1.67'-01	-9.90'-02	1.00'-02	7.54'+04
	10.00								
10		1.34'+07	1.80'+08	3.08'+06	7.43'+01	1.52'-01	-8.90'-02	1.00'-02	6.57'+04
	11.00								
11		9.93'+06	1.43'+08	2.29'+06	6.51'+01	1.38'-01	-7.90'-02	1.00'-02	5.62'+04
	12.00								
12		7.17'+06	1.11'+08	1.66'+06	5.65'+01	1.24'-01	-7.00'-02	1.00'-02	4.72'+04
	13.00								
13		5.02'+06	8.51'+07	1.17'+06	4.85'+01	1.10'-01	-6.00'-02	1.00'-02	3.86'+04
	14.00								
14		3.39'+06	6.33'+07	7.95'+05	4.11'+01	9.70'-02	-5.00'-02	1.00'-02	3.07'+04
	15.00								
15		2.18'+06	4.56'+07	5.17'+05	3.42'+01	8.30'-02	-3.90'-02	1.00'-02	2.34'+04
	16.00								
16		1.34'+06	3.15'+07	3.19'+05	2.79'+01	7.00'-02	-2.80'-02	1.00'-02	1.70'+04
	17.00								
17		7.56'+05	2.06'+07	1.82'+05	2.22'+01	5.80'-02	-1.70'-02	1.00'-02	1.14'+04
	18.00								
18		3.86'+05	1.25'+07	9.50'+04	1.71'+01	4.60'-02	-3.00'-03	1.00'-02	6.74'+03
	19.00								
19		1.69'+05	6.76'+06	4.32'+04	1.23'+01	3.60'-02	1.20'-02	1.00'-02	2.91'+03
	20.00								

Figure 1. Program: CBEAM 4, version: January 15, 1981

Nibe B, wooden rotor blade, data 3, 12 mm plywood, mod.: E-module
No. of elements: N = 19. Rotation-omega = 3.50 rad/sec.
Conicity = 0.105 rad. Pitch angle = -0.262 rad.
Eccentricity of ref.-axis = 0.00 m.

Fig 1

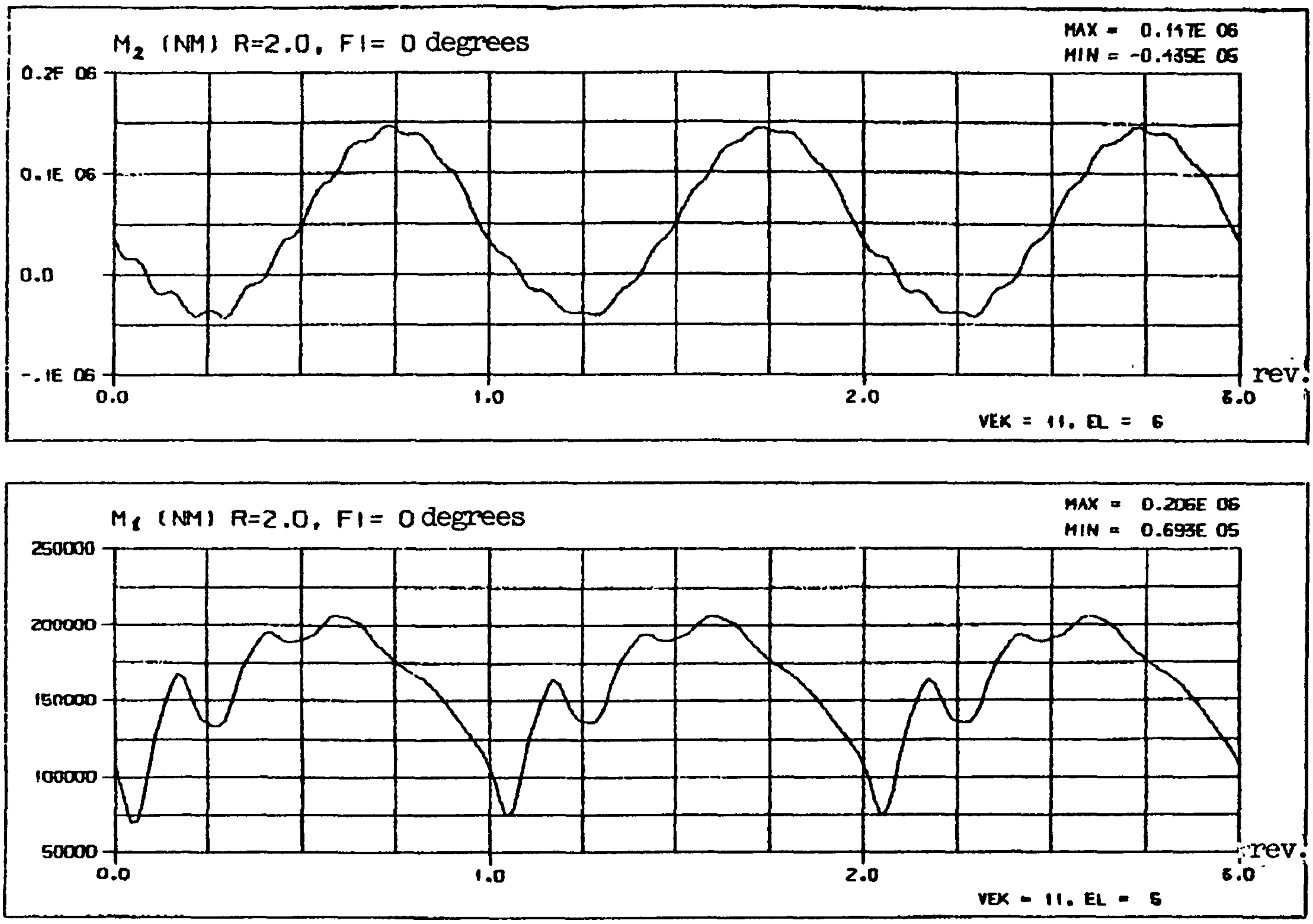
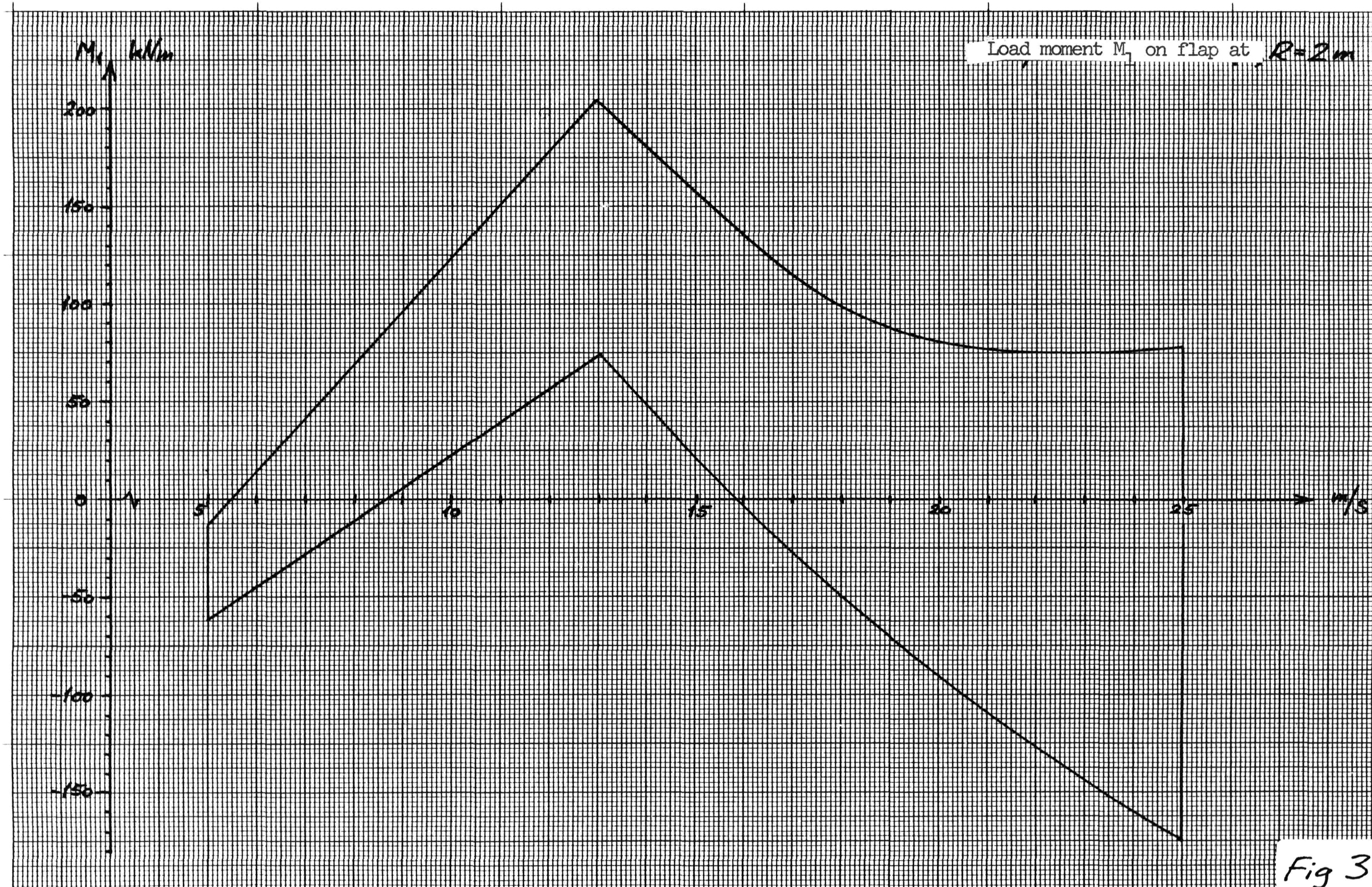
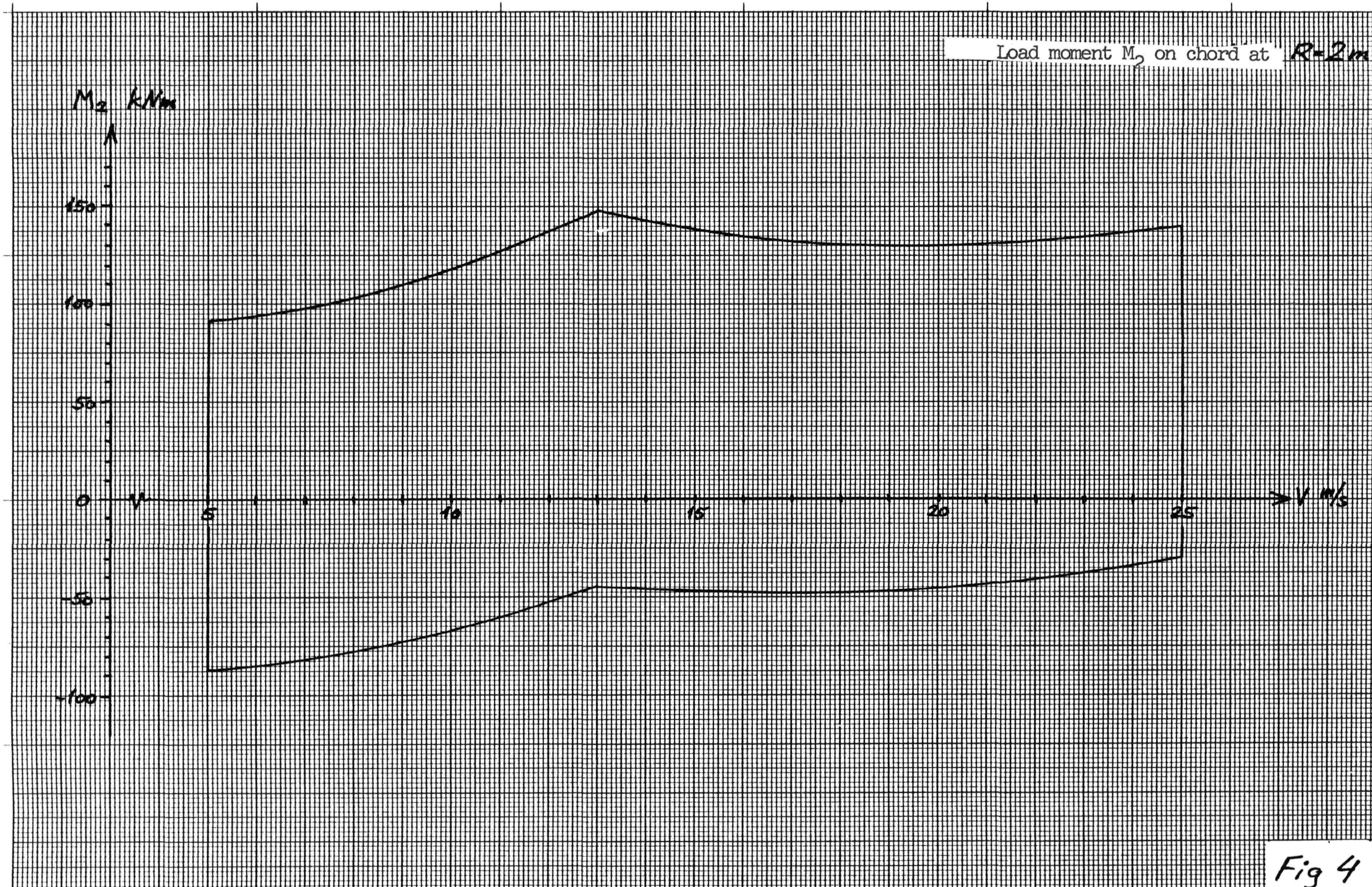


Figure 2. Wind = 13 m/sec, $\Theta p = 0^{\circ}$, yaw = $+15^{\circ}$



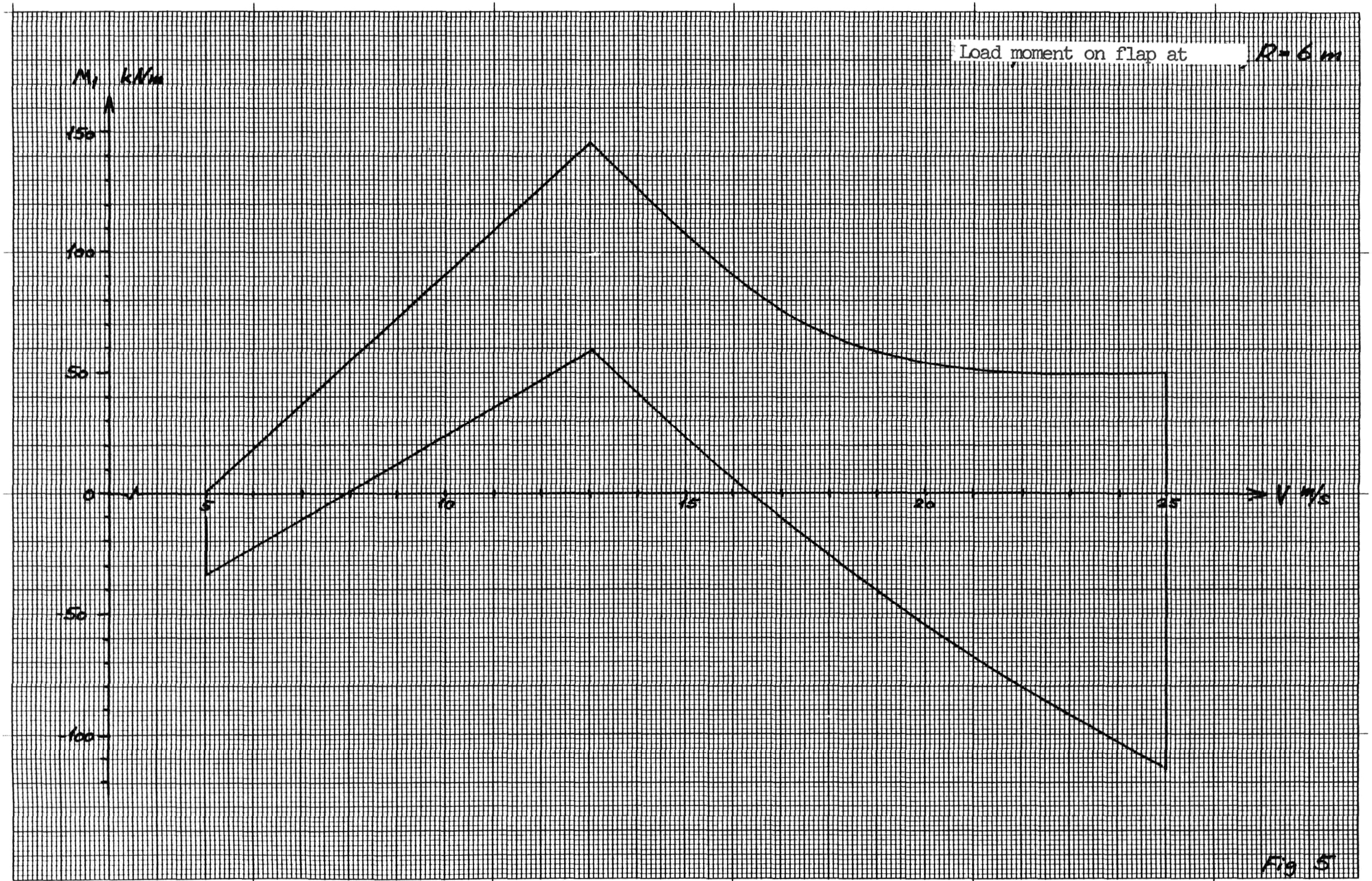
Load moment M_1 on flap at R=2m
M_1 kNm
200
150
100
50
0
-50
-100
-150
5
10
15
20
25
m/s

Fig 3



Load moment M_2 on chord at R=2m
M_2 kNm
150
100
50
0
-50
-100
V
5
10
15
20
25
V m/s

Fig 4



Load moment on flap at R = 6 m
M_1 kNm
150
100
50
0
-50
-100
V m/s
5
10
15
20
25

Fig 5

Load moment M_2 on chord at R = 6m

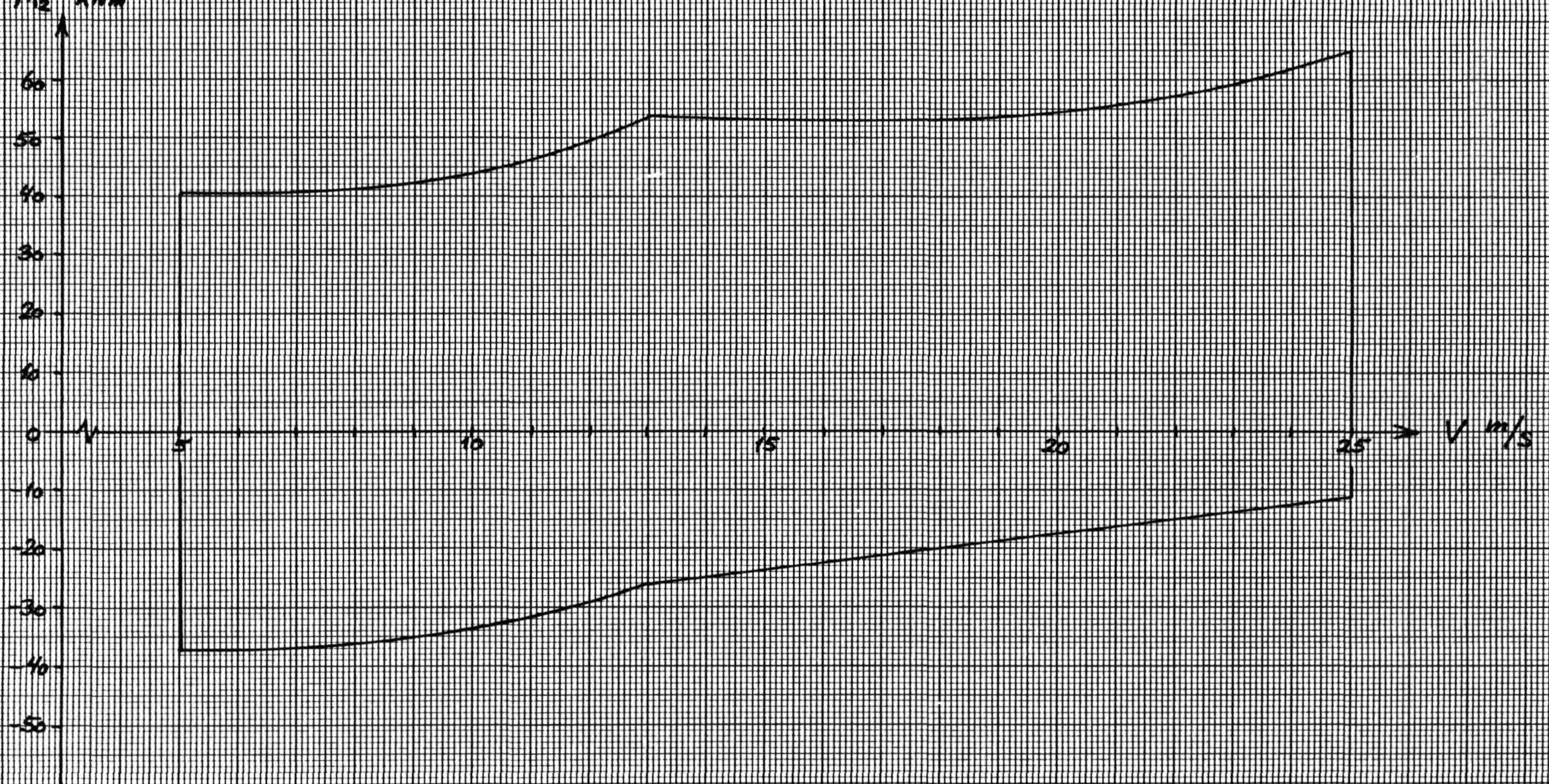
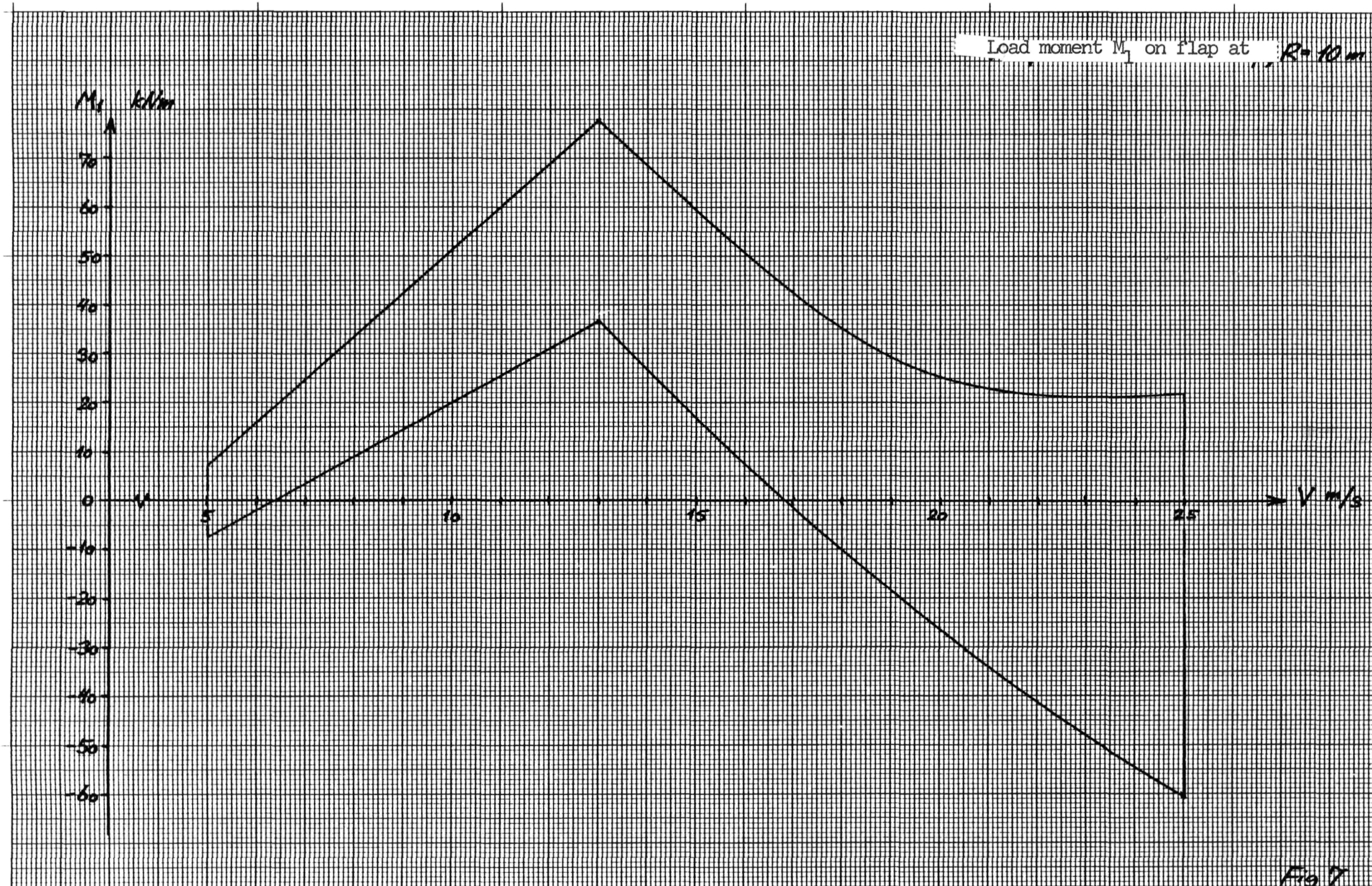
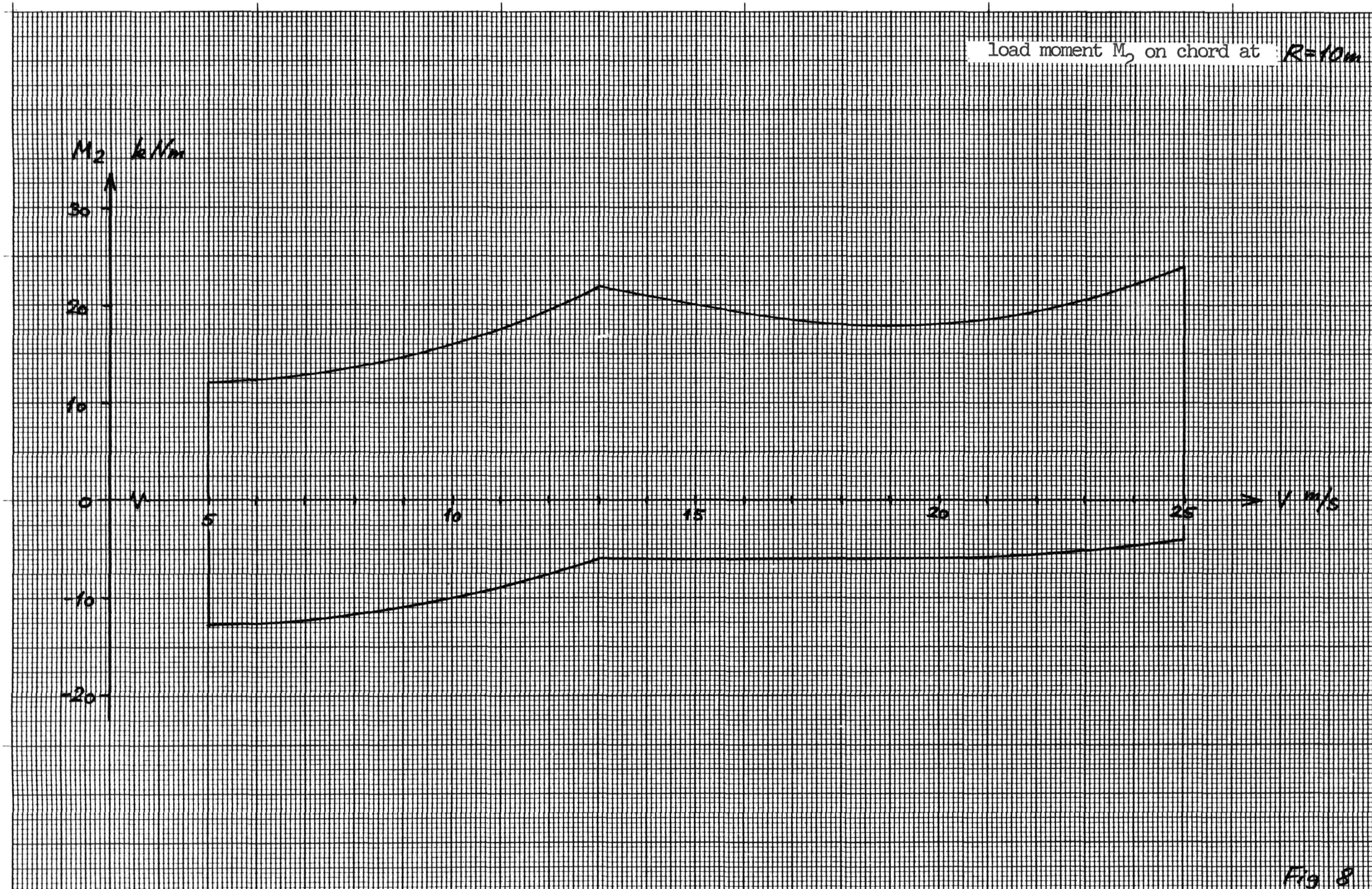


Fig 6



Load moment M_1 on flap at R = 10 m
M_1 kNm
70
60
50
40
30
20
10
0
-10
-20
-30
-40
-50
-60
V
5
10
15
20
25
V m/s

Fig 7



load moment M_2 on chord at $R=10m$
M_2 kNm
30
20
10
0
-10
-20
5
10
15
20
25
V m/s

Fig 8

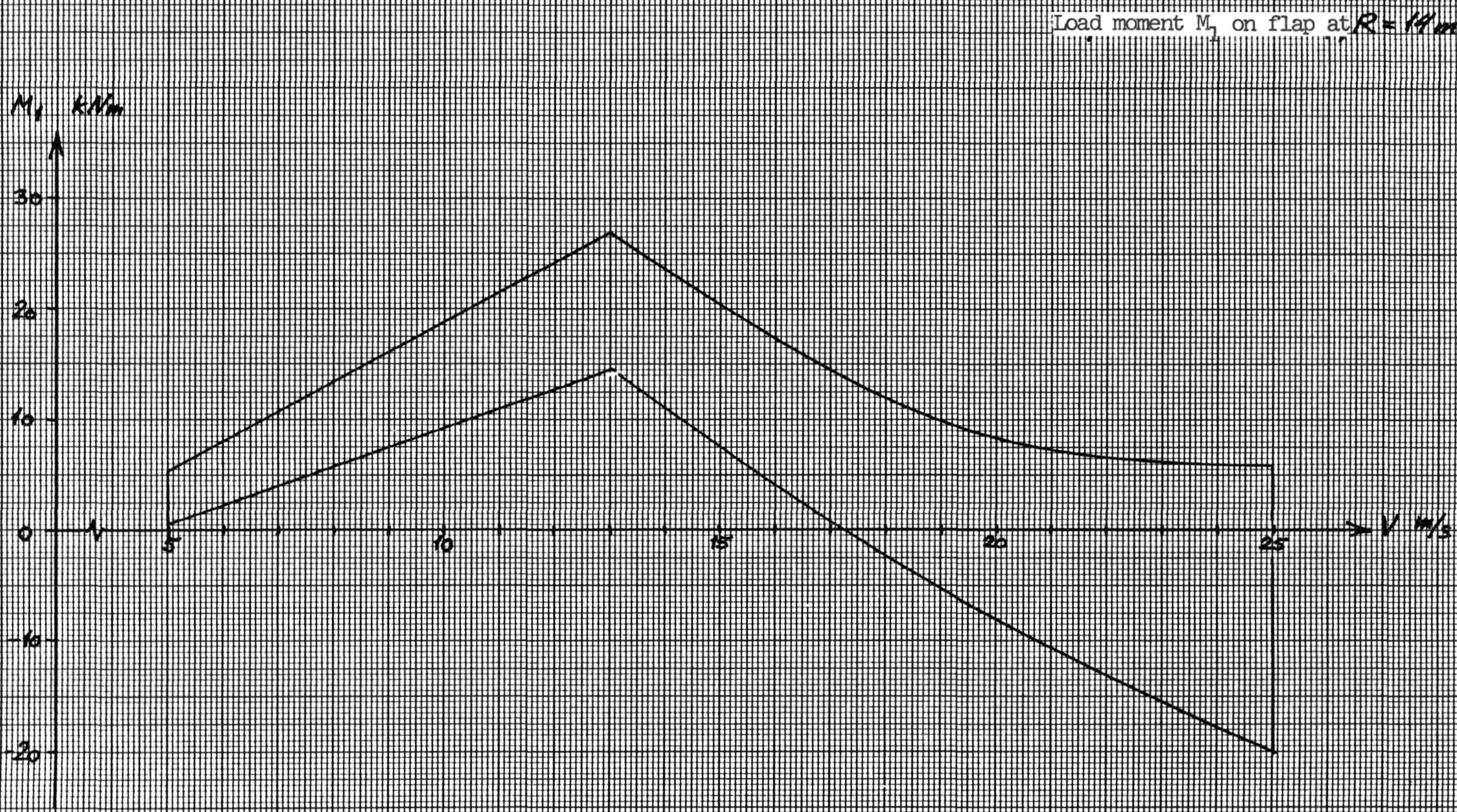
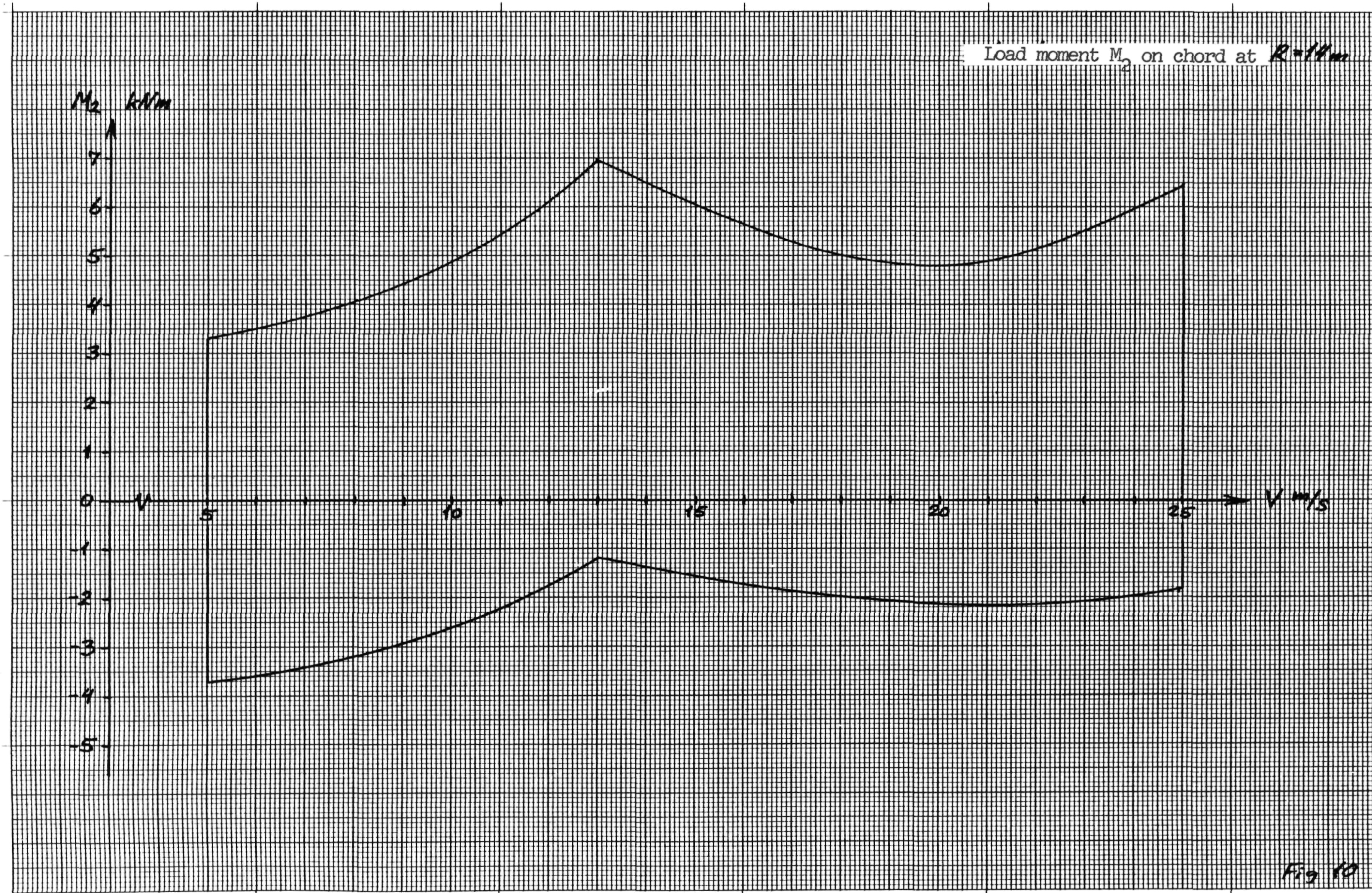

Load moment M_l on flap at R = 14 m
M_l kNm
30
20
10
0
-10
-20
V m/s
5
10
15
20
25

Fig 9



Load moment M_2 on chord at R=14m
M_2 kNm
7
6
5
4
3
2
1
0
-1
-2
-3
-4
-5
V
5
10
15
20
25
V m/s

Fig 10